The Development of Locomotive Power at Speed

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The power developed per unit of cylinder volume of locomotives in the speed range from 250 to 400 r.p.m. has in recent years been doubled. Forty years ago it was often the case that the diminution of the mean effective pressure with increase of speed, rather than the evaporative capacity of the boiler, limited the power of the engine. The most advanced designs maintain the mean pressure at a high percentage of the calculated value to the highest speeds, and the effect of valve events and clearance volume on the calculated mean pressure thus becomes of practical importance, especially as designers are endeavouring to use considerably higher steam pressures in single-expansion cylinders.

The first part of this paper sets out the effect of these factors over a wide range of steam pressures in the form of basic data graphs of mean pressure, relative efficiency, and steam consumption. It is shown that without the decrease of clearance volume and increase of expansion ratio which compound expansion affords, the improved thermal efficiency which higher steam pressure offers cannot be fully realized above about 250 lb. per sq. in. boiler pressure, though greater power can be obtained.

The second part of the paper is devoted to an examination of the actual deviation of mean pressure with speed for a wide range of locomotives, including some of the most recent designs, and the reasons for the radical improvement are indicated. In 1905 the late Professor Dalby suggested a simple proportional relationship between mean pressure and speed. Such a relationship does not hold for modern locomotives, and at the end of the paper a simple exponential law is proposed with a single coefficient characteristic of the locomotive. This law may be used to estimate the power of a future design at any speed, or as a criterion for assessing the performance of an existing locomotive in respect of power developed, which is generally of more importance to railway companies than thermal efficiency.

Introduction. Steam locomotive design may appear to have remained static for the last fifty years as compared with the technical developments of stationary steam plants and other forms of prime mover. It is true that there have been no fundamental changes in the construction or the working principle of the steam locomotive apart from the use of higher steam temperatures. Yet the power capacity of the average locomotive of 1900 was a mere fraction of that of the best contemporary designs, for the same unit cylinder volume *at the speed of running prevailing in 1939*. If the performance of locomotives over the range of speed 250–400 r.p.m. is examined, it may justly be claimed that progress has been commensurate with that of other established types of engine. If it had not been so, it is almost certain that the non-condensing reciprocating engine would not have maintained its pre-eminence for railway service.

It has to be admitted that the spectacular developments of locomotive performance in recent years are not entirely the result of devices or principles unknown at the beginning of the century. The performance of a locomotive in practice is, however, so dependent on the steam-producing capacity of the boiler that there grew up a habit of basing any study of its power on the evaporative capacity of the boiler. Some years ago the author made a survey† of the various methods that have been used for calculating and measuring locomotive horse-power, and it will be found that almost all the early formulae were based on the boiler, notably those of Frank (1887) and Goss (1901) which expressed the power of a locomotive as a function of the heating surface. It has generally been a fundamental assumption that the power of a locomotive is limited by its evaporative capacity, but whilst this is obviously true in a broad theoretical sense, it has not always been true in practice of large-wheeled locomotives burning good coal. The author has had experience of such locomotives of older design which when in good running order would steam freely at any speed in full gear and at full regulator opening, the power being limited only by the capacity of the engine to take steam at speed and to exhaust it without excessive rejection of energy.

General appreciation of the fact that the remarkable capacity of the locomotive boiler to generate steam surpassed the ability of the engine to make use of the steam, was also delayed by the complications arising from the absence of a normal running condition in locomotive operation. In stationary steam plant practice the engines or turbines are designed to run at a specified speed and ratio of expansion, and no doubt has existed that the logical procedure is to fix the capacity of the plant as the power of the turbines at the designed operating conditions, the designer being responsible for seeing that sufficient boiler capacity is provided to supply the steam required. There are no such fixed design operating conditions in the case of a locomotive. Moreover the fact that before the advent of the stationary test plant, locomotive tests were generally carried out under fluctuating conditions, made it difficult to determine the effect of speed on the power and efficiency of the engine.

The introduction of the test plant, notably that of the Pennsylvania Railroad at Altoona, and of constant-speed methods of road testing, focused attention on the great variation in locomotive performance in the higher speed range and gave a powerful impetus to the improvements in the design of valve gears and steam circuit which have produced such a transformation of the high-speed performance of the locomotive.

The mean effective pressure in a locomotive cylinder can, of course, be readily calculated for any given steam conditions, valve events, and clearance, assuming only the back pressure during exhaust. The calculated mean effective pressure is independent of the speed so that the indicated horse-power derived from it would be directly proportional to the speed. In practice the mean effective pressure falls as the speed increases, and it is this increasing deviation from the ideal which may limit the power of the locomotive just as effectively as the

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† An alphabetical list of references is given in the Appendix, p. 416.

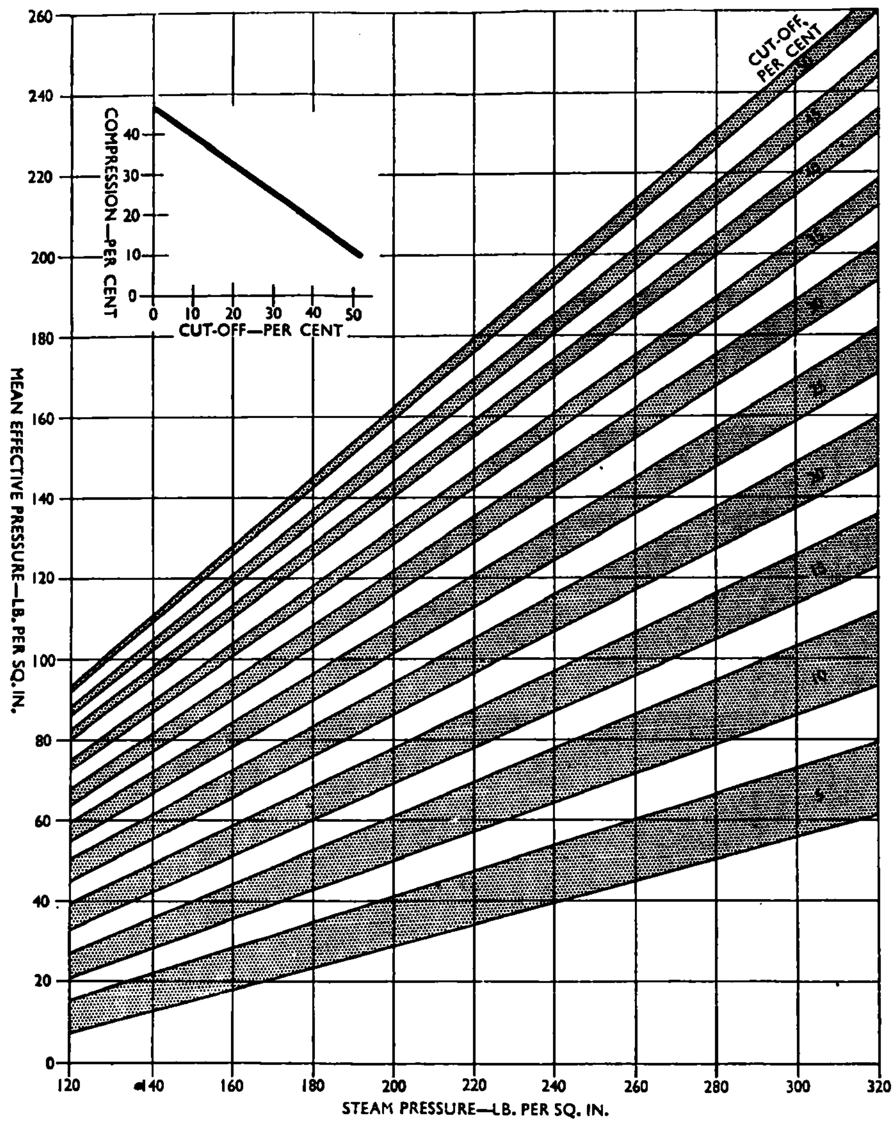


Fig. 1. Calculated Mean Effective Pressures for Various Steam Pressures and Cut-offs (Clearances 10–15 per cent)

Basis: Exhaust pressure, 18 lb. per sq. in. abs.; compression point as given by inset graph; law of expansion, $PV^{1\cdot 33}$=const.; law of compression, $PV^{1\cdot 2}$ = const.

Each shaded band gives the mean effective pressures corresponding to the cut-off indicated, for percentage clearance spaces of 10 per cent (lower line) to 15 per cent (upper line). Values for intermediate clearances may be obtained by interpolation.

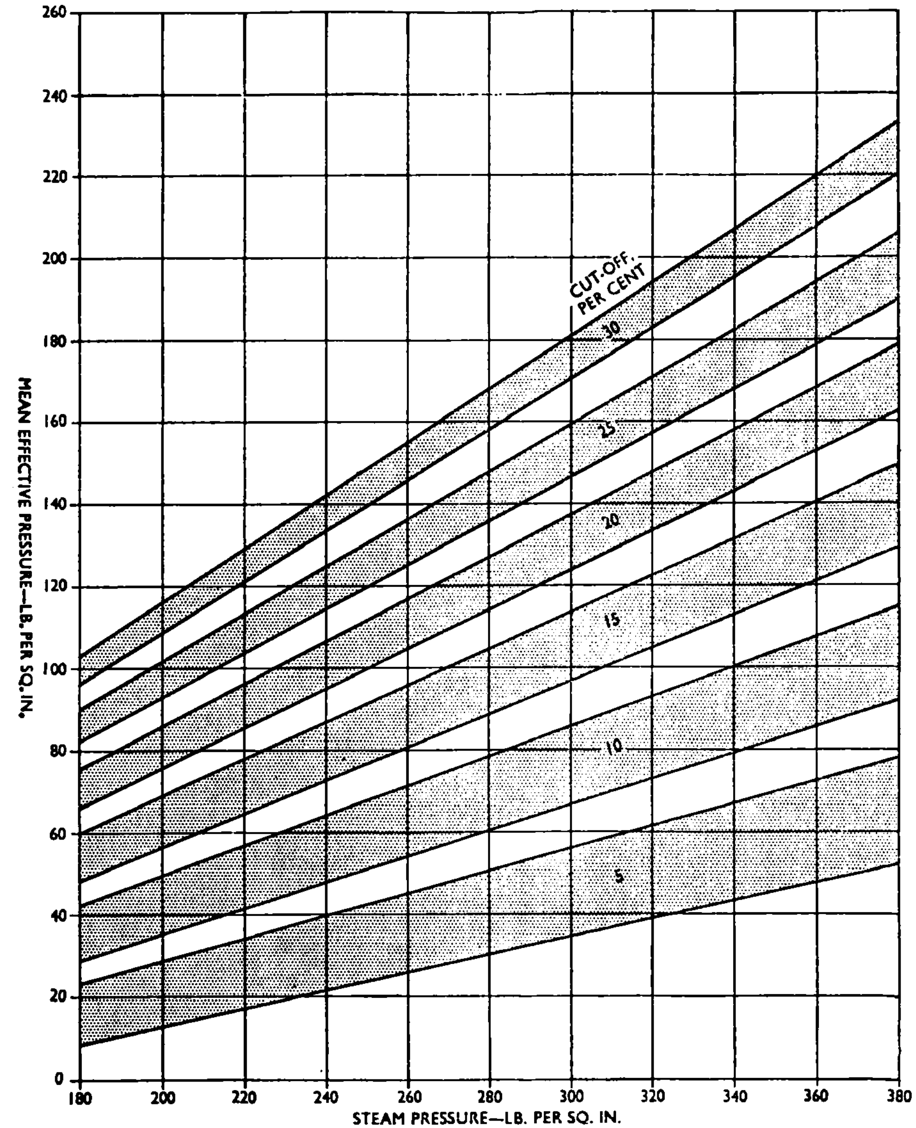


Fig. 2. Calculated Mean Effective Pressures for Various Steam Pressures and Cut-offs (Clearances 5–10 per cent)

Basis as for Fig. 1. The lower line of each shaded band gives the mean effective pressure for 5 per cent clearance, and the upper line for 10 per cent clearance.

capacity of the boiler. The late Professor Dalby (1905) proposed an equation for the relationship of mean effective pressure and speed based on a simple proportion with coefficients peculiar to each engine. Dalby's mean effective pressure curve is shown in Fig. 9, p. 410, and it will be seen that it is a straight line cutting the zero ordinate at a speed almost within the range 250–400 r.p.m. Such a locomotive would be unable to move itself on the level at a speed of 80 m.p.h. no matter how big its boiler or how large its cylinders. To-day a locomotive has been built capable of hauling a train of 1,000 tons weight at a speed of 100 m.p.h. on the level*.

Though Dalby's equation was of doubtful validity over the higher speed range, it did indicate in a general way the state of locomotive design of that day, and the attenuation of the indicator diagram at express speed was so great and so variable as to render any conclusions to be drawn from calculated mean effective pressures of purely academic interest. To-day, the rate of deviation of the mean effective pressure with increase of speed still varies widely and is the most important characteristic of the engine, but in the best designs it has been reduced to such an extent that an analysis, such as is attempted in the first part of this paper, of the relationship between mean effective pressure and the boiler pressure, cut-off, and clearance, together with the corresponding cylinder efficiencies, becomes of great practical importance, especially in designing locomotives for the relatively higher pressures that are now being adopted.

BASIC LOCOMOTIVE DATA GRAPHS

Owing to the wide and variable deviations from the theoretical indicator diagram which in the past were characteristic of locomotive performance, such broad data as were generally accepted were based simply on the boiler pressure. The best known is the universally accepted tractive effort formula which assumes that the greatest pressure that can be exerted in the cylinders is 0·85 of the boiler pressure. Even comparatively modern formulae for power such as those of Cole (1914), Kiesel† (1915), and Lipetz (1933) are based on factors unrelated to cut-off or clearance. Cole employed empirical speed factors by which the percentage of the maximum tractive effort available at each speed is estimated; Kiesel used an expansion ratio which is not the indicated cut-off, but the ratio of the weight of a cylinder-full of steam to the weight of steam supplied per stroke on the basis of the assumed evaporation rate; while Lipetz's method was derived from Goss's formula and was based primarily on the evaporative power of the boiler.

It is hoped to show in this paper that by basing curves of mean effective pressure over a range of speed on the calculated mean effective pressure for the appropriate conditions, as was done by Dalby, a more exact expression for power can be deduced which, by the use of an appropriate coefficient, can be universally applied to locomotives of widely differing types. Such a method is more in accord with fundamental theory and for this reason alone is less dependent on empirical relationships which may hold only for a particular phase or type of locomotive design. Its value is greatly enhanced by the fact that modern valve gears give more precise and finer control over the expansion ratio than was formerly the case.

If, moreover, the actual mean effective pressure at speed is commensurate with the calculated mean pressure, as is the case in the most advanced designs, then a number of conclusions of great importance in designing locomotives for higher steam pressures can be drawn from such calculated data, as will be shown in the following paragraphs.

Relationship of Mean Effective Pressure to Steam Pressure and Cut-off. Fig. 1 is the basic data sheet showing calculated mean effective pressures for all steam pressures from 120 lb. per sq. in. to 320 lb. per sq. in. and over the whole range of economical cut-offs, from 5 per cent to 50 per cent. It also gives the data for percentage clearance spaces from 10 to 15 per cent. To avoid overlapping, data for clearances down to 5 per cent are given in Fig. 2, and as these are normally only applicable to compound locomotives, they are only given for cut-offs up to 30 per cent and for the higher range of steam pressures. Mean pressures for intermediate clearance percentages can be obtained within the general limits of accuracy of the graphs by linear interpolation‡.

The main assumption necessary in making these calculations relates to the point of compression. The inset curve in Fig. 1 gives the relationship between cut-off and compression used. Radial valve gears give a relationship of this nature, and for practical reasons such a relationship would be desirable with any type of gear. It gives maximum efficiency at early cut-offs, which correspond to normal running at speed, and it sacrifices efficiency to extra power at the late cut-offs used for starting or at slow speeds. Thus it is also the ideal relationship on theoretical grounds. The curve is typical of a modern Walschaert gear. It also gives results which are not appreciably affected in the case of a compound locomotive with two sets of gears, each operating at a late cut-off, if the equivalent overall cut-off is used in reading the total mean effective pressure from the main graph.

The other assumptions relate to the exhaust pressure and the laws of expansion and compression. The exhaust pressure has been fixed at a constant pressure of 18 lb. per sq. in. abs. This is about the minimum attainable in practice at speed. With increasing power it is bound to rise, of course, but that is only part of the general attenuation of the indicator diagram for which these curves are intended to provide a basic standard of comparison. The law of adiabatic expansion of superheated steam is taken to be $PV^{1 \cdot 33}$ = const.§ Theoretically the steam at the end of expansion is generally a little below saturation point, but in practice as a result of throttling it is usually slightly above saturation. In this instance it has been thought advisable to make a compromise between theory and practice and a law of compression $PV^{1 \cdot 2}$ = const. has been assumed. The effect on the calculated mean effective pressure is in any case small.

Effect of Clearance on Mean Effective Pressure. It will be observed that at the earlier cut-offs necessary for economical working in the higher range of pressure, the percentage of clearance space has a very considerable effect. For modern locomotives working at a high expansion ratio this effect cannot be neglected in any calculations of power based on cylinder dimensions. It is of importance to know also the effect of clearance on the steam action in the cylinder as otherwise the increase in power may be illusory.

Relative Cylinder Efficiency at Increased Pressure. The calculation of the relative efficiencies corresponding to the curves in Figs. 1 and 2 is a simple matter. The weight of steam remaining in the clearance space at the point of compression can be calculated from its volume (clearance + compression), its pressure, and its state. To avoid the introduction of any arbitrary element its state is taken strictly as that given by steam tables after adiabatic expansion between the pressures concerned. The steam supplied per stroke is then the amount of steam at boiler conditions less the amount of steam trapped at compression. It is necessary only to know the steam temperature at admission. In practice this varies greatly, even in the same locomotive with the rate of evaporation. For the purpose of a basic comparison, however, a single steam temperature may be taken for all conditions, and a mean steam temperature of 600 deg. F. (315 deg. C.) has been selected. The relative efficiency is the amount of work done per pound of steam at the calculated mean effective pressure compared with the work done on the Rankine cycle by a pound of steam in the same state, expanding from the boiler pressure to the exhaust pressure. The efficiency of the Rankine cycle itself varies according to these limiting pressures, and this is taken into account in Fig. 6.

* The four-cylinder "Duplex" locomotive built for the Pennsylvania Railroad by the Baldwin Locomotive Works, described in a paper to the New York Railroad Club in May 1945 by R. P. Johnson, Chief Engineer of the Baldwin Locomotive Company. See p. 414 and Fig. 19.

† Published by A. J. Wood (1925).

‡ It should be explained that a high degree of accuracy is not necessary for any of the purposes contemplated in this paper, and all the calculations have been performed by a 10-inch slide rule and 4-figure logarithms.

§ Schüle, W. "Technical Thermodynamics", pp. 380–1. All the data in this paper are valid only for superheated steam.

In Fig. 3 the relative efficiencies are plotted over the same range of boiler pressures and cut-offs as in Fig. 1, for 5, 10, and 15 per cent clearance respectively. The drop in the early cut-off curves at low boiler pressures is due to the expansion lines falling below the exhaust lines in these extreme conditions. These curves show that with the same clearance it is necessary for a locomotive with a boiler pressure of 250 lb. per sq. in. to be worked at effective cut-offs from 20 to 30 per cent of the stroke shorter than a locomotive with a boiler pressure of 180 lb. per sq. in. in order to maintain the same relative cylinder efficiency. Put in another way, if a locomotive with a steam pressure of 250 lb. per sq. in. is normally worked at the same effective cut-off as a locomotive with a pressure of 180 lb. per sq. in., say, 15 per cent cut-off at full speed, the relative cylinder efficiency

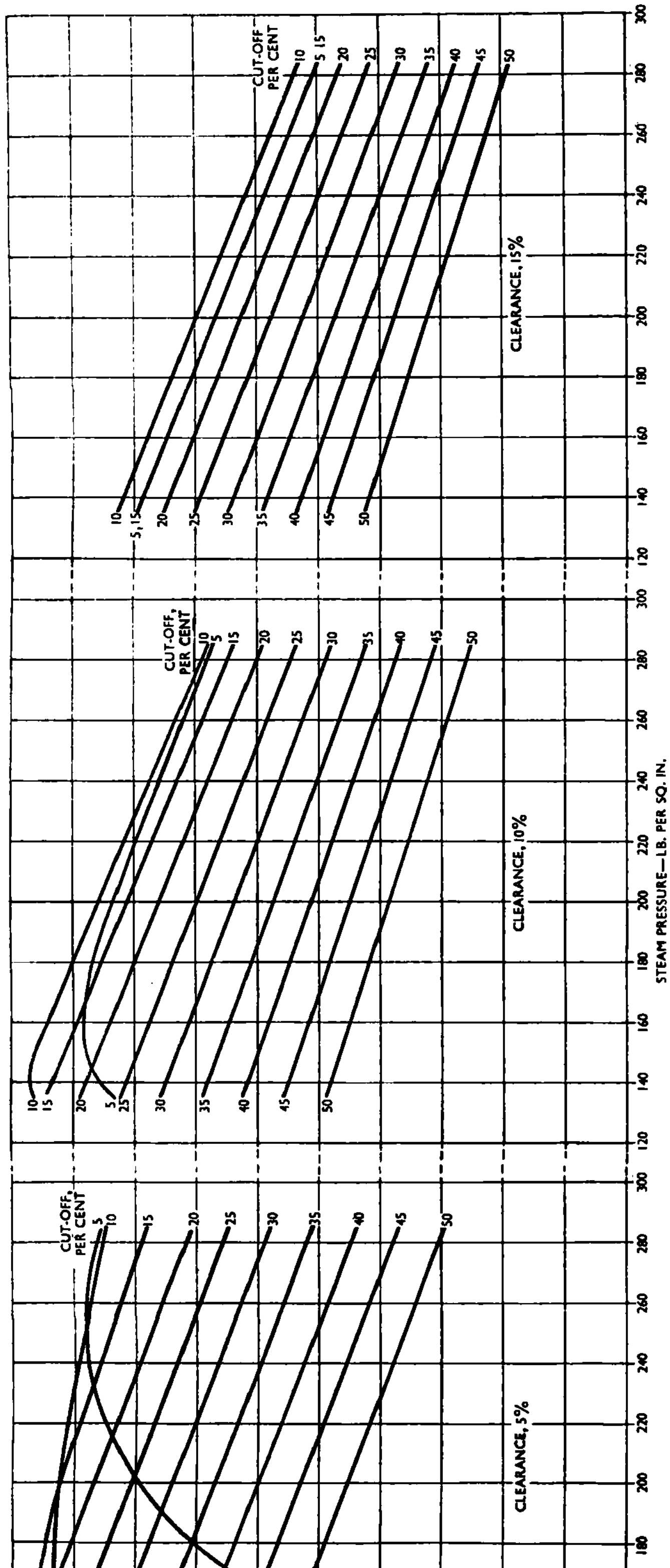


Fig. 3. Relative Cylinder Efficiencies for Various Steam Pressures and Cut-offs

Basis: Mean effective pressure as for Figs. 1 and 2. Steam temperature at admission, 600 deg. F.

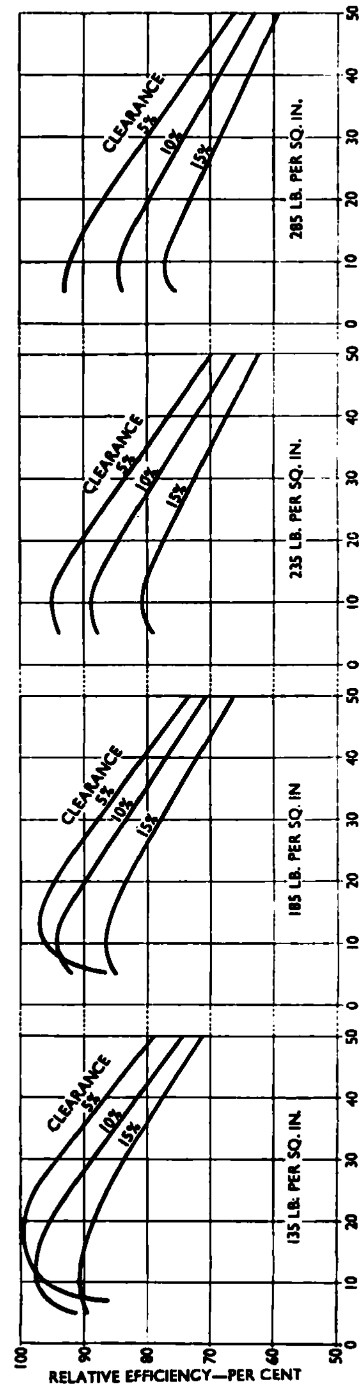


Fig. 4. Relative Cylinder Efficiencies on a basis of Cut-off

Basis: Mean effective pressure as for Figs. 1 and 2. Steam temperature at admission, 600 deg. F.

The three curves for each steam pressure represent 5 per cent clearance (top curve), 10 per cent clearance (middle curve), and 15 per cent clearance (bottom curve) respectively.

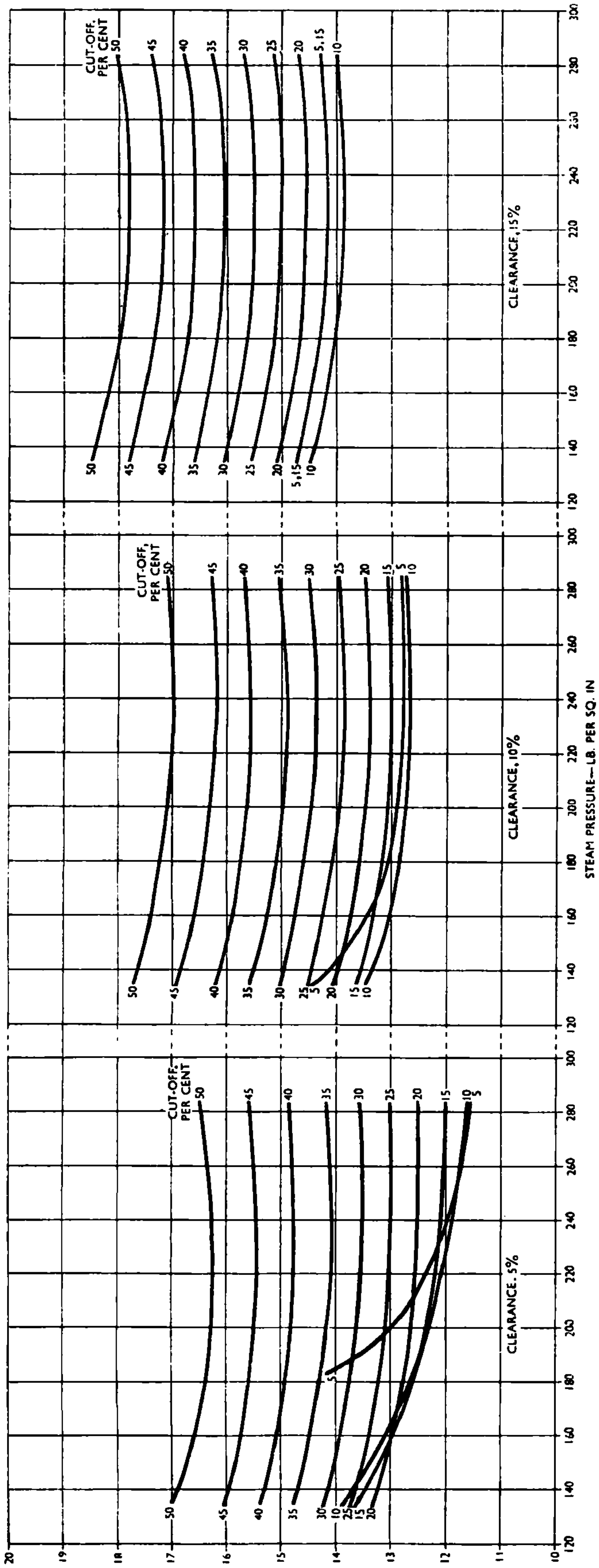


Fig. 6. Specific Steam Consumptions for Various Steam Pressures and Cut-offs

will be 7 per cent less. As the Rankine cycle* efficiencies for the two pressures are 20·3 and 18·4 per cent respectively, this means that the effect of the reduced expansion ratio nearly nullifies the gain in efficiency due to the higher pressure, the respective cylinder thermal efficiencies being 17·3 and 17·0 per cent.

Effect of Clearance on Relative Efficiency at Increased Pressure. Fig. 4 shows the effect of clearance in another way, on a basis of cut-off. It will be seen that owing to the effect of clearance it is theoretically impossible under normal running conditions at speed to realize the higher cylinder thermal efficiencies associated

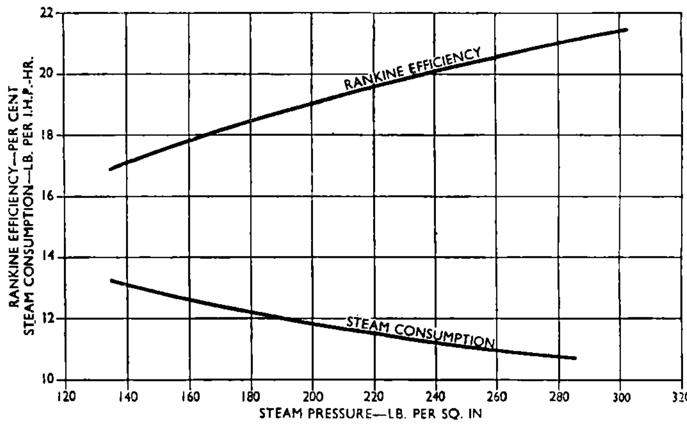


Fig. 5. Rankine Efficiency and Corresponding Steam Consumption for Various Steam Pressures

BASIS: Exhaust pressure, 18 lb. per sq. in. abs.; steam temperature at admission, 600 deg. F. Available heat reckoned above temperature of exhaust steam.

with higher steam pressures as compared with locomotives of lower boiler pressure equipped with modern valve gears permitting early cut-off working. The increase of cylinder thermal efficiency (Rankine cycle) for the same terminal conditions as for the other diagrams and the corresponding decrease in steam consumption, are plotted in Fig. 5. It should be said that boiler pressures have been increased primarily in order to increase the power obtainable within the limits of the prescribed loading gauge and axle weight. But, of course, any limitation of efficiency is likewise a limitation of power.

Fig. 6 completes the basic data by giving the steam consumption per indicated horse-power-hour over the full range of steam pressures and cut-offs for 5, 10, and 15 per cent clearance respectively. These graphs present the net effect of the changes in opposite directions of the Rankine efficiency and the relative efficiency respectively.

Nordmann's High-Pressure Locomotive Tests. Professor Nordmann has reported (1938) a systematic series of constant-speed road tests carried out on the German State Railways to ascertain the saving in steam consumption of a number of locomotives built with a boiler pressure of 294 lb. per sq. in., together with an experimental compound locomotive with a pressure of 368 lb. per sq. in. Two of the locomotives were also tested at a reduced pressure of 206 lb. per sq. in. to provide a direct comparison. The results are summarized in Fig. 7 and Table 1. It will be seen from Fig. 7 that the experimental compound locomotive gave results which stand quite apart, especially in the speed range above 250 r.p.m. For the simple-expansion engines there is no overriding saving in steam consumption.

* The Rankine efficiency is throughout reckoned in this paper as the adiabatic heat drop between the boiler pressure and the exhaust pressure (18 lb. per sq. in. abs.) divided by the heat available above the liquid temperature corresponding to the exhaust pressure. It is true that in locomotives the feed water temperature is generally nearer the temperature of the atmosphere than that of the exhaust steam, but partial feed water heating is frequently practised. This does not, of course, affect the values of the relative cylinder efficiencies.

TABLE 1. COMPARATIVE STEAM CONSUMPTIONS OF TWO SIMILAR 2–10–2 TANK LOCOMOTIVES AT TWO DIFFERENT BOILER PRESSURES

The calculated steam consumptions are those given by Fig. 6 for the same *nominal* cut-off.

Locomotive	Speed, r.p.m.	206 lb. per sq. in.					294 lb. per sq. in.				
		Cut-off, per cent	I.h.p.	Calculated steam consumption, lb. per i.h.p.-hr.	Actual steam consumption, lb. per i.h.p.-hr.	Relative efficiency (Rankine cycle), per cent	Cut-off, per cent	I.h.p.	Calculated steam consumption, lb. per i.h.p.-hr.	Actual steam consumption, lb. per i.h.p.-hr.	Relative efficiency (Rankine cycle), per cent
2-cylinder	114	50	1,610	17·1	16·4	71·3	37	1,890	15·2	14·0	75·7
	152	46	1,830	16·3	14·5	81·0	34	2,090	15·0	12·7	83·7
	228	40	1,940	15·6	13·6	86·0	28	2,070	14·3	12·8	83·0
3-cylinder	114	56	1,529	17·8	17·4	67·4	38	1,741	15·5	15·2	69·7
	152	49	1,768	16·9	15·0	78·3	30	1,970	14·6	13·7	77·5
	228	41	1,811	15·7	14·6	80·4	23	1,895	13·8	14·2	74·5

Table 1 sets out the comparative results for the two locomotives which ran at two pressures. These locomotives were of similar type except that one had two cylinders and the other three. The clearance volume in both cases was approximately 10 per cent. At the lower speed at which late cut-offs were used an improvement in relative efficiency was observed, but at the higher speed there was a decrease of relative efficiency despite the earlier cut-offs at the higher pressure. The reduction in steam consumption on an indicated horse-power-hour basis was reported to be insufficient to compensate for the reduction in mechanical efficiency which was found to accompany the higher pressures, so that the net result was actually a higher steam consumption per drawbar horse-power-hour at the higher speeds. There was considerable variation in the percentage clearance as between the other locomotive types tested, and Nordmann attributed the variable results largely to the effect of clearance.

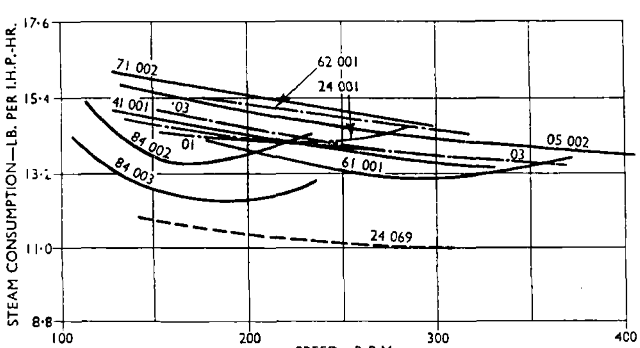


Fig. 7. Steam Consumption of Various Locomotives with High Boiler Pressures (Nordmann)

———— 294 lb. per sq. in.
- - - - - - Experimental compound locomotive (368 lb. per sq. in.).
—·—·— 235 or 206 lb. per sq. in.

The numbers are those of individual locomotives tested. The two-cylinder and three-cylinder tank locomotives referred to in Table 1 were numbered (confusingly) 84 003 and 84 002 respectively.

In comparing these results with the basic data it should be remembered, of course, that this section of the paper deals with theoretical indicator diagrams only. The actual steam consumptions recorded were generally appreciably lower than those given by the basic data (see Fig. 6), due principally to the fact that at the late cut-offs employed throttling at admission is considerable and the effective cut-off was therefore considerably earlier than the nominal. But in their general relationship to each other the results, which constitute the most comprehensive investigation of the effect of increased boiler pressure on the actual performance of locomotives which has yet been published, do demonstrate the value of basic data graphs as a general guide to the locomotive designer.

Advantages of Compound Expansion in obtaining Increased Cylinder Efficiency with Increased Steam Pressure. The relative efficiency of the calculated or perfect diagram is greatly affected by the clearance volume, the exhaust pressure, and the point of compression, because these determine the additional quantity of live steam required to fill the clearance volume at admission pressure. This quantity expands partly without doing external work, and the earlier the cut-off the greater the resultant percentage loss. There comes a point when the gain due to an increase in the ratio of expansion is exceeded by this relatively increasing clearance loss. This is the reason for the upward trend of the steam consumption curves in Fig. 6 above 250 lb. per sq. in.

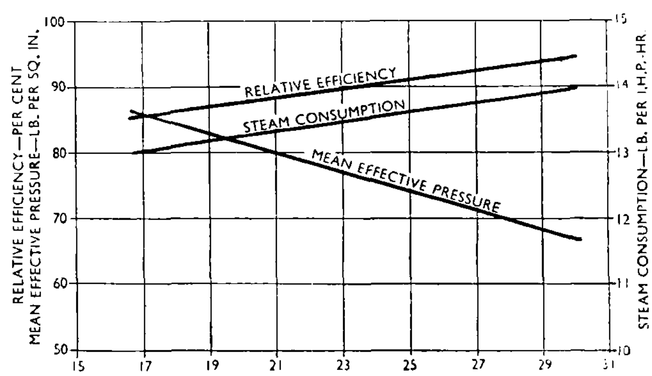


Fig. 8. Effect of Variation of Exhaust Pressure on Relative Efficiency and Specific Steam Consumption

Steam pressure, 235 lb. per sq. in.; cut-off, 15 per cent; clearance, 10 per cent.

In practice, with imperfect steam circuit and valve gear, the exhaust pressure tends to rise well above the assumed basic pressure at speed, and in consequence a greater weight of steam is trapped and compressed in the clearance space. With imperfect engines it may for this reason be possible to secure a continuous improvement in the relative cylinder efficiency with increase of pressure, even at early cut-offs. The Rankine efficiency, however, decreases at a greater rate with increase of back pressure and the net result is an increase in the specific steam consumption. The effect of back pressure for a typical set of conditions is exhibited in Fig. 8.

Appreciable departures of the relationship between cut-off and compression from that shown in the inset to Fig. 1 will have a greater effect on the efficiency and steam consumption than on the mean effective pressure. A greater percentage of compression is equivalent to a lower percentage clearance, and if it were possible to design the valve gear so as to compress the clearance steam to admission pressure the ideal would be attained and the steam consumption curves would fall continuously with increase of steam pressure.

The difficulty, however, is that locomotives are controlled by throttling when very limited power is required, and under these conditions such a gear would give excessive compression and knocking. As things are, it is a very common practice for drivers to lengthen the cut-off when working with a low steam chest pressure—a practice which is not illogical as is shown by the curves of relative efficiency.

For the highest efficiency at steam pressures above about 250 lb. per sq. in., therefore, compound expansion would appear to be essential. Apart from the fact that it enables the ratio of expansion to be increased in normal running sufficiently to keep pace with the greater working range of pressure, there is the extremely important advantage that only the clearance volume of the high-pressure cylinder affects the basic relative efficiency, provided, as will be explained later, the correct proportions between the cylinders *and their clearance volumes* are chosen.

There are, of course, many vital practical considerations which may be of more importance to the designer than the attainment of the highest efficiency, which it is beyond the scope of this paper to enter into. It should also be emphasized that high-pressure locomotives run for a large proportion of their time at a reduced steam chest pressure, the full pressure constituting a reserve for maximum efforts. A full appreciation of the conditions governing efficiency will, however, assist in forming a correct assessment of the conflicting factors which the designer has to balance so carefully.

EFFECT OF SPEED ON MEAN EFFECTIVE PRESSURE

Mean Effective Pressure-Speed Curves for L.M.S. Railway Locomotives, 1917–1944. It has already been explained that the remarkable development of locomotive performance in recent years is due to the maintenance of a higher proportion of the

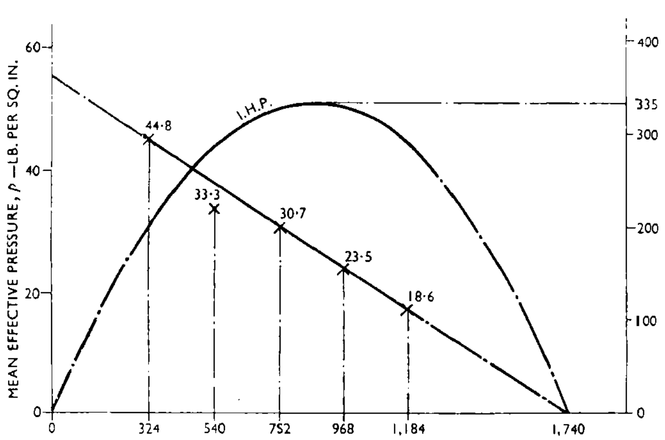


Fig. 9. Early Curve of Mean Effective Pressure (Dalby)

Steam pressure, 130 lb. per sq. in.; cut-off, 25 per cent.

attainable mean effective pressure at high speed. Fig. 9 shows Professor Dalby's mean effective pressure curve of 1905 for the locomotive *Schenectady* tested on Professor Goss's pioneer stationary testing plant at Purdue University. The associated curve of indicated horse-power is also shown.

Fig. 10 shows a curve of mean effective pressure for a typical Midland Railway superheater express locomotive of 1917, and Fig. 11 is plotted from a series of indicator diagrams taken in 1944 by the London, Midland and Scottish Railway on a rebuilt *Royal Scot* locomotive incorporating Walschaerts valve gear representative of the best steam distribution attained with this type of gear.

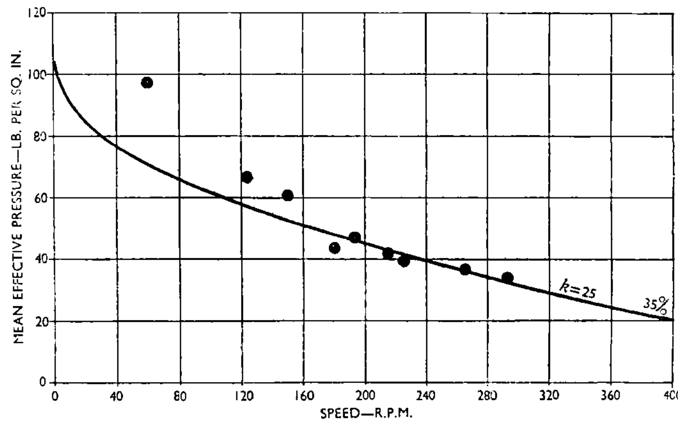


Fig. 10. Mean Effective Pressure Curve for Typical Superheater Express Locomotive of 1917

Steam pressure, 160 lb. per sq. in.; cut-off 35 per cent; clearance approximately 10 per cent.

*Characteristic power equation: $P_N = P_C\,(1 - \frac{1}{25}\sqrt{N})$.

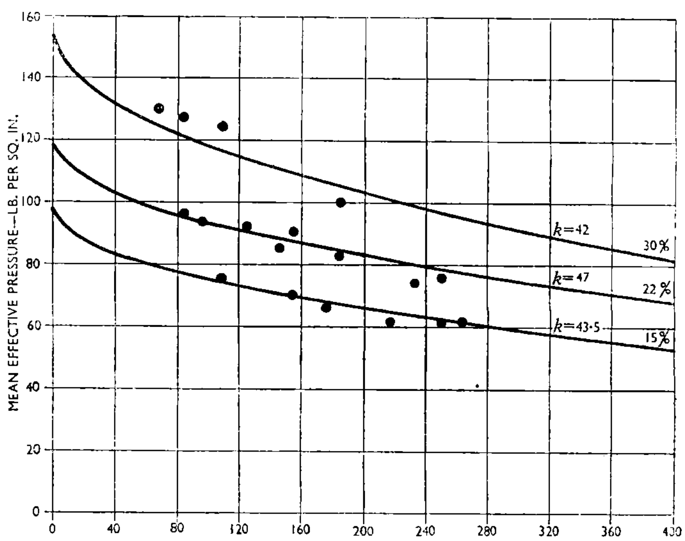


Fig. 11. Mean Effective Pressure Curves for Rebuilt *Royal Scot* Locomotive (1944)

Steam pressure, 250 lb. per sq. in.; clearance approximately 10 per cent.

Characteristic power equation: $P_N = P_C\,(1 - \frac{1}{44}\sqrt{N})$.

It will be observed that instead of a straight line as in Dalby's diagram, exponential curves have been drawn through the plotted points. The basis of these curves is considered in the final section of the paper (p. 415), but in the meantime it is important to note that in all cases they meet the zero speed line at the basic mean effective pressure obtained from Figs. 1 or 2. Comparison of the L.M.S. curves shows that at 300 r.p.m. the percentage of the basic mean effective pressure developed by the two locomotives is:—

4-4-0 superheater locomotive (1917) . .	31 per cent
Rebuilt *Royal Scot* locomotive (1944) .	61 „

* See p. 415.

This means that at a road speed of approximately 72 m.p.h. the power developed per unit cylinder volume has been increased by practically 100 per cent.

The curves for both locomotives were plotted from indicator diagrams taken during ordinary running. Those for the *Royal Scot* engine were all at full regulator opening. The diagrams for the Midland engine were reported as being taken at the same setting of the reversing gear, but examination of the cards shows that the cut-offs at low speed were considerably later than the nominal value, whereas the cards at the higher speeds agree well with the stated cut-off. This accounts largely for the displacement of the points below 150 r.p.m. Regarding these and the succeeding curves, apart from uncertainty as to the exact point of cut-off, other major causes of scatter of the points are variation of superheat temperature, which is normally of the order of 100 deg. F. for the same locomotive under various conditions of running, and variations of the amount of leakage.

Reasons for Improvement in Performance at Speed. Considerations of space will not permit of a detailed discussion of the improvements in valve gear design which have contributed to these results, and in any case they have been extensively discussed elsewhere*. In his 1905 paper Dalby drew attention quite explicitly to the causes of the extreme attenuation of the diagrams which he studied. The only locomotive engineer in this country who at that time was following the same line of thought was the late G. J. Churchward, whose outstanding genius as a designer is perhaps only now being fully recognized. The author has been told by Sir William Stanier that Churchward set up a single locomotive cylinder and engine unit at Swindon with which he tried out a number of modifications to valve gears. The author has previously suggested that valuable research could be accomplished by this means without the necessity of a full-scale testing plant, and was unaware that Churchward had, in fact, satisfied himself as to the great advantage of increasing the steam lap of the valve by such means. A recent example of the value of such a method of testing is the successful development of the Franklin system of poppet valve gear on a laboratory test plant at Baltimore, U.S.A. (Alcock 1944).

In his Presidential Address† Sir William Stanier pointed out that the advent of superheating delayed the general improvement of valve gears and steam circuit because of the greater inherent fluidity of superheated steam; this considerably reduced the losses due to throttling and high back pressure. That there was nevertheless still ample scope for improvement is shown by Fig. 12 in which actual indicator diagrams at high speed for the two L.M.S. locomotives are compared with the calculated basic diagrams.

The adoption of the piston valve greatly facilitated the improvement of steam ports: it was no longer necessary for them to be close together in the steam chest. It has, however, brought problems of leakage with it which have at times threatened to undo the good it has effected, and have necessitated the overcoming of many technical problems in design.

The increase of the steam lap of the valve has the consequence of increasing the amplitude of its harmonic motion for the same valve events. Theoretically, by increasing the lap, and hence the amplitude, sufficiently it would be possible to obtain full port opening at admission at early cut-offs, and extremely sharp opening and closure of the valve at exhaust, but here again it is the overcoming of the practical problems associated with the inertia of the valve and valve gear that determines the progress that can be made.

Table 2 gives particulars of the valve gears of the two locomotives to which Figs. 10–12 apply, so that they may be related to the results achieved.

Mean Effective Pressure Curves for Chapelon's Compound Locomotive Reconstructions. In 1936 M. André Chapelon of the French National Railways published his monumental paper entitled "Les Progrès Récents de la Locomotive à Vapeur". The Paris-Orléans Company built some Pacific type compound

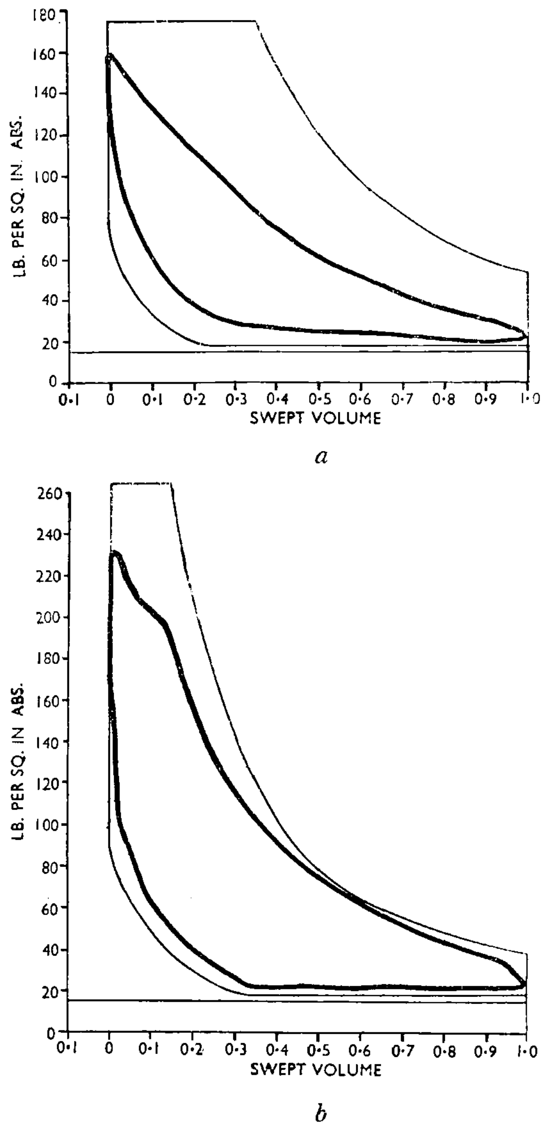


Fig. 12. Comparison of Basic and Actual Indicator Diagrams

a 4–4–0 superheater locomotive (1917).
b Rebuilt *Royal Scot* locomotive (1944).

locomotives in 1909 which, like many locomotive designs of the day, gave much less power at high speed than was thermodynamically possible—in this instance 1,850 i.h.p. instead of over 3,000 i.h.p., as estimated at a later date. It was decided to rebuild these engines, and the opportunity was taken to reconstruct a number of individual machines experimentally with larger steam passages and chests and different types of valve

TABLE 2. PARTICULARS OF VALVE GEARS OF L.M.S. LOCOMOTIVES

	Midland 4–4–0	Rebuilt *Royal Scot*
Type of valve gear . . .	Stephenson	Walschaerts
Steam lap of valve . . .	1 inch	$1\frac{3}{8}$ inches
Width of steam port . . .	$1\frac{1}{2}$ inches	$2\frac{1}{4}$ inches
Exhaust clearance . . .	Nil	$\frac{1}{16}$ inch
Lead (full fore gear) . . .	$\frac{3}{16}$ inch	} $\frac{5}{16}$ inch
,, (mid gear) . . .	$\frac{9}{32}$ inch	
Piston valve diameter . .	8 inches	9 inches
Maximum valve travel . .	$3\frac{3}{4}$ inches	6 inches

* See, for instance, Sanford (1931) and Windle (1931).
† Proc. I.Mech.E., 1941, vol. 146, p. 50.

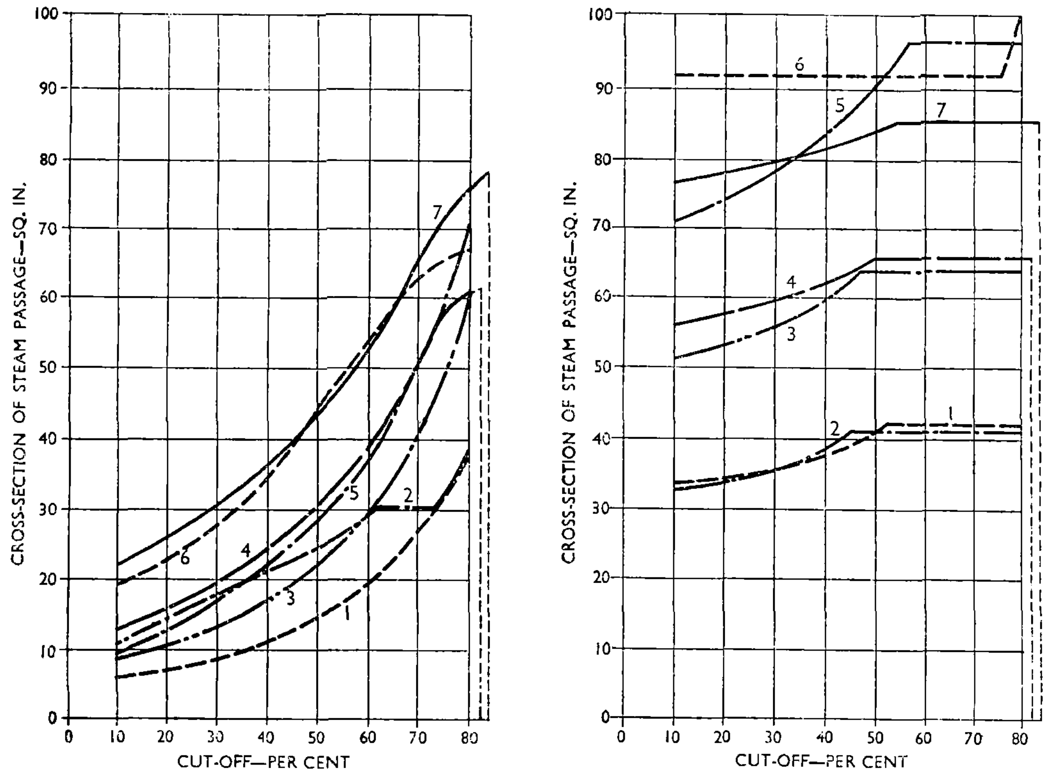


Fig. 13. Cross-sectional Areas of Steam Passages at Admission and Exhaust given by Different Types of Valves Tried (Chapelon)

1 Plain slide valves.
2 Trick valves with double admission.
3 Piston valves.
4 Poppet valves.
5 Piston valves with extra long travel (7½ inches).
6 Willoteaux piston valves with double admission and double exhaust.
7 Poppet valves of enlarged type.

gear. As the necessity for replacements arose, various rebuilds also incorporated other improvements, including a new form of superheater giving a higher degree of superheat, various form of Kylchap double exhaust, the Nicholson syphon, and a higher boiler pressure. So continuous a rebuilding programme resulted in almost every combination of improvements being embodied in different locomotives, and Chapelon's paper gave a full account of the results achieved, during constant-speed road tests, by each stage of development.

It was, perhaps, the most systematic piece of experimental work in the history of locomotive design, and the successive improvements in the steam circuit and valve gear are particularly relevant to the subject of the present paper. The data were not presented in a form which enabled the mean effective pressure curves to be plotted readily, and the author is greatly indebted to M. Chapelon who, despite the tragic dislocation of French locomotive developments in the last five years, has kindly searched out the records and supplied the author with the data from which the curves in Figs. 14 and 15 have been constructed.

Chapelon's work constitutes a stage of development beyond that represented by the L.M.S. curves, in that two successive types of balanced poppet valve were used, actuated by oscillating cams driven by Walschaerts gear. Poppet valves have, of course, been tried in this country, but it has been found generally that improvements to piston valves and gears have given equally good results. The poppet valve, however, is also capable of development. Fig. 13, which is reproduced from Chapelon's paper (English dimensions and designations being substituted), shows the geometrical cross-sectional area of the steam passage at admission and exhaust given by every type of valve tried, for various cut-offs, and it will be seen that the piston valve with 7½ inches maximum travel gave a better opening than the original type of poppet valve, but that the second type of poppet valve gave the best openings.

As regards the steam chests and steam pipes, it is pleasant to record that M. Chapelon recalls an old rule of that great English pioneer D. K. Clark, whose paper on "Expansive Working of Steam in Locomotives" read before this Institution in 1852 can still be read with interest. Clark laid down that the volume of the high-pressure steam chests should be equal to that of one cylinder. This rule had long been forgotten till the Nord Company proved once more its value in some tests in 1897. It was decided to double the cross-section of the steam passages, by increasing the ratio of their cross-section to the area of the pistons from $\frac{1}{10}$ to $\frac{1}{5}$.

Before dealing with the mean effective pressure curves some additional comment is necessary on the application of the basic data graphs to compound engines. For the purpose of ascertaining the equivalent cut-off the proportion of the volume of the high-pressure cylinder filled during admission is related to the swept volume of the low-pressure cylinder in the usual way. The mean effective pressures are likewise referred to the low-pressure cylinder. Similarly the clearance volume is taken as a percentage of the low-pressure cylinder volume, since the low-pressure clearance does not affect the quantity of live steam admitted per stroke. If there is any loss due to the low-pressure clearance it appears as a loss of mean effective pressure. But in locomotives the range of pressure in the low-pressure cylinder is such that it is not difficult to arrange that compression is practically complete. It is, however, important that, having predetermined accurately the receiver pressure for normal conditions, the correct ratio of the low-pressure clearance to the high-pressure be determined to ensure the correct degree of compression in the low-pressure cylinder.

Turning now to the curves, Fig. 14 represents a direct comparison of the original form of this compound locomotive, No. 3579 (1909) with the engine (No. 3705) rebuilt with poppet valves of the first type on both high- and low-pressure cylinders. In comparing No. 3579 with the Midland simple-expansion locomotive it has to be remembered that each set of valve gear in the compound works at a much later cut-off for the same overall ratio of expansion, and consequently the severe throttling associated with short-lap valves is less apparent. It will be observed that the points for the later cut-offs depart appreciably from the curves at low speed and would conform better to Dalby's straight-line law. This is sometimes the case with engines of early design, as might be expected, and reference is made to this on p. 415.

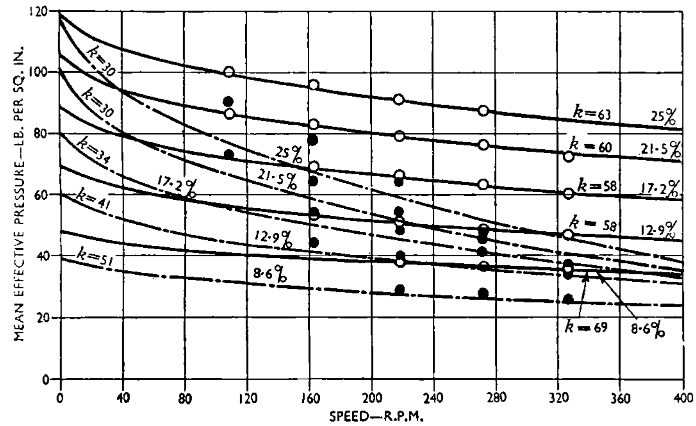


Fig. 14. Mean Effective Pressure Curves for Original and Rebuilt Paris-Orléans Compound Locomotives

—●— Original locomotive No. 3579: Steam pressure, 228 lb. per sq. in. Characteristic power equation: $P_N = P_C\,(1-\frac{1}{37}\sqrt{N})$.

—○— Rebuilt locomotive No. 3705: Steam pressure, 242 lb. per sq. in. Characteristic power equation: $P_N = P_C\,(1-\frac{1}{61}\sqrt{N})$.

The result of the combination of compound expansion with poppet valves on high- and low-pressure cylinders, as well as the enlargement of the cross-sections of steam chests and passages, is shown by these curves to be a close approach to the maximum attainable power for given steam conditions and cut-off. Even at 400 r.p.m. (90 m.p.h.) the mean effective pressure developed is about 68 per cent of the basic calculated figure, at all cut-offs.

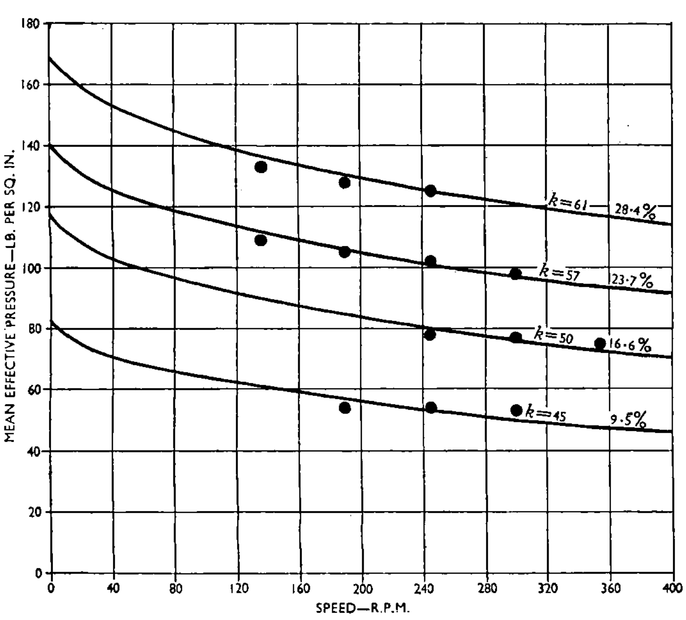


Fig. 15. Mean Effective Pressure Curves for Rebuilt Compound Locomotive with High Boiler Pressure and Poppet Valves of Enlarged Type (Locomotive No. 4701)

Steam pressure, 284 lb. per sq. in.
Characteristic power equation: $P_N = P_C(1-\frac{1}{34}\sqrt{N})$.

Fig. 15 represents a series of reconstructions with a considerably higher boiler pressure (284 lb. per sq. in.) and poppet valves of the second (enlarged) type. In this diagram the curves show a performance rather inferior to those in Fig. 16, though the actual points plotted give an indication of an even more level series of mean pressures. The reason appears to lie in the excessive clearance space with the enlarged valve gear, amounting to no less than 25 per cent of the high-pressure cylinder volume, as compared with 13·2 per cent for No. 3579 and 16·4 per cent for No. 3705. (The virtual clearance, referred to the low-pressure cylinder, for locomotives Nos. 3579, 3705, and 4701, is 5·7, 7·0, and 11·8 per cent respectively.) This also accounts for the fact that the deterioration increases as the cut-off is shortened. It seems probable from the slope of the plotted points that if the clearance volume could have been kept to a normal value with the enlarged poppet valves and steam passages, this locomotive would have shown the best characteristic power equation of the whole series in this paper.

M. Chapelon, in his letter to the author, evaluated these curves by expressing the diminution of the ratio (α) of the mean effective pressure to the steam chest pressure, between 120 and 320 r.p.m., as a percentage of the mean effective pressure at 120 r.p.m., and he compiled the table reproduced here as Table 3. Such a method, while it gives a correct evaluation of the rate of deviation of the mean pressure over a particular range of speed when the deviation approximates to a straight line, takes no account of losses of pressure between the boiler and the steam chest, loss of heat to the cylinder walls, piston leakage, clearance loss, or any other losses which may have a constant element, and, as we have seen, can in consequence be misleading.

TABLE 3. COMPARISON OF PERCENTAGES OF STEAM CHEST PRESSURE INDICATED AT 120 AND 320 R.P.M. BY VARIOUS LOCOMOTIVES (CHAPELON)

Locomotive No.	3579	3705	4701	5399
Virtual cut-off, per cent	21·5	21·5	23·7	22·0
Percentage of steam chest pressure indicated at 120 r.p.m.	32·5	36·0	39·0	36·5
320 „	19·0	31·8	35·5	29·0
Diminution of α or indicated mean pressure between 120 and 320 r.p.m. per cent	41·5	11·6	9·0	20·5

The locomotive No. 5399, referred to in the last column of Table 3, is a "Pacific" of the K4S type of the Pennsylvania Railroad, for which mean effective pressure curves are given in Fig. 18. This locomotive was rebuilt with poppet valves on the Franklin system and a detailed account of it was given by Alcock (1944). The reconstruction gave the following increases in the cross-sections of the steam passages:—

Admission (at 25 per cent cut-off)	100 per cent
Exhaust „ „ „	4–9 „
Ports	32 „

M. Chapelon comments in his letter, on Table 3, as follows: "While the diminution of the ratio α or of the tractive effort with speed between 120 and 320 r.p.m. reaches 41·5 per cent for the original P.O. Pacific locomotive, it is only 11·6 and 9 per cent respectively for the modified locomotives and this diminution still amounts to 20·5 per cent in the American single-expansion engine, despite the benefit of a much-improved steam circuit.

"Contrary to the general opinion which may have been correct formerly, the losses in passing from the high-pressure to the low-pressure cylinders are thus very far from handicapping the compound as compared with the simple-expansion engine from the point of view of the tractive effort realizable at speed."

Altoona Test Plant Curves of Mean Effective Pressure. In Figs. 16–18 similar sets of curves are presented for the express passenger locomotives tested on the Altoona plant of the Pennsylvania Railroad. They are all simple-expansion engines with

Walschaerts valve gear operating long-lap piston valves. It will be observed that the curves fit the plotted points well except in the case of the K4S with piston valves.

This locomotive, though similar in general dimensions to the K2SA and E6S locomotives, was deliberately built with considerably enlarged cylinders, with the intention of getting the same power at an earlier and therefore more economical cut-off.

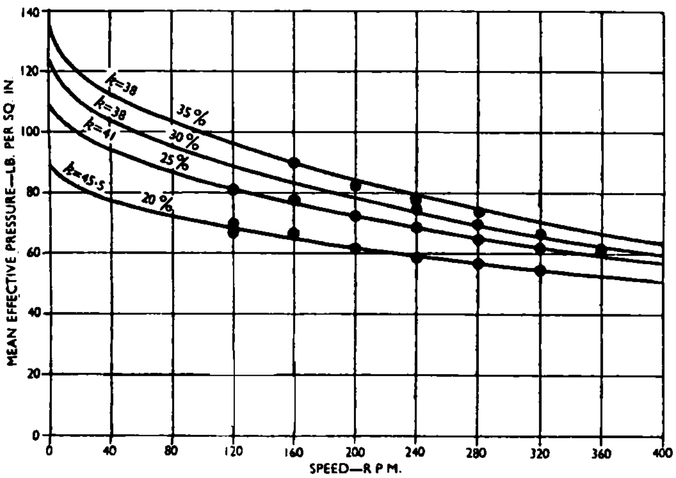


Fig. 16. Mean Effective Pressure Curves for K2SA Locomotive (Altoona Tests)

Steam pressure, 205 lb. per sq. in.; clearance, 13·5 per cent.
Characteristic power equation: $P_N = P_C\,(1-\frac{1}{41}\sqrt{N})$.

The steam passages, however, were not increased proportionately to the increase in cylinder volume and there was in consequence a considerably increased drop in pressure between the boiler and the cylinders during admission, especially through the superheater, for the same cut-off. This did not affect the thermal efficiency of the engine as the valve gear was reasonably efficient and there was no excessive loss at exhaust

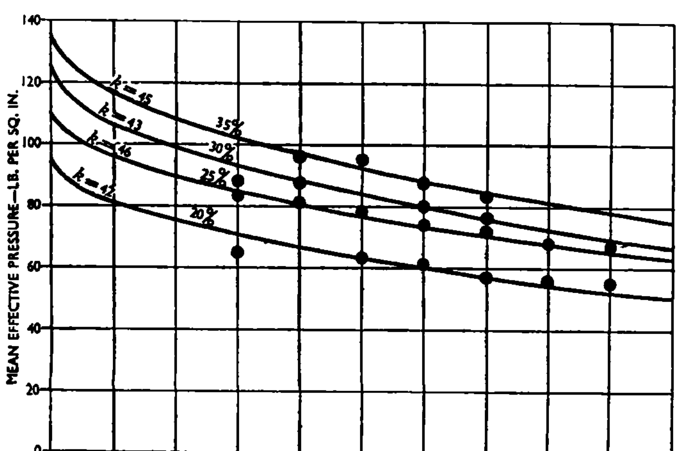


Fig. 17. Mean Effective Pressure Curves for E6S Locomotive (Altoona Tests)

Steam pressure, 205 lb. per sq. in.; clearance, 14 per cent.
Characteristic power equation: $P_N = P_C\,(1-\frac{1}{44}\sqrt{N})$.

at high speed. In fact the increase in economy for any given power, due to the reduced clearance (10 per cent as compared with 14 per cent) and the greater expansion ratio, was generally greater than the loss, giving a net improvement of efficiency. The plotted points deviate from the curves in the manner that would be expected in these circumstances, as is explained on p. 415.

Mean Effective Pressure–Speed Curves for Simple-Expansion Locomotives with Franklin Poppet Valve Gear. As the boiler pressure and general dimensions of the locomotive remained unchanged, the curves for the reconstructed K4S locomotive

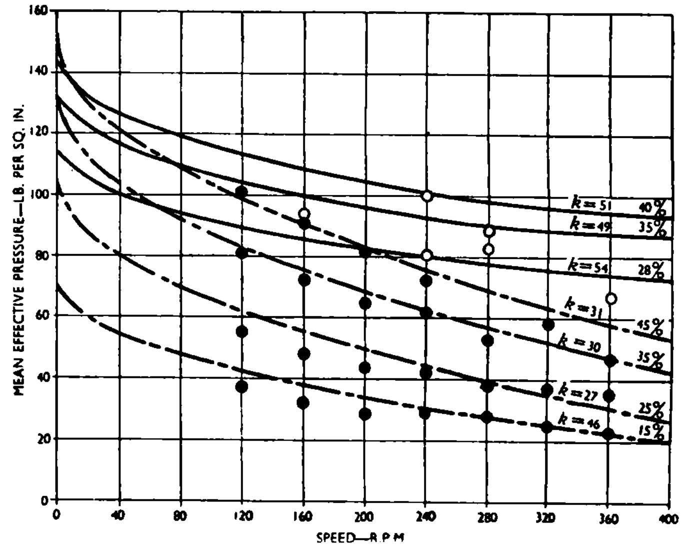


Fig. 18. Mean Effective Pressure Curves for K4S Locomotive with Piston Valves and with Franklin Poppet Valves and Enlarged Steam Passages (Altoona Tests)

Steam pressure, 205 lb. per sq. in.; clearance, 10 per cent.
- —●— - Original locomotive: Characteristic power equation: $P_N = P_C\,(1-\frac{1}{34}\sqrt{N})$.
—○— Rebuilt locomotive No. 5399: Characteristic power equation: $P_N = P_C\,(1-\frac{1}{52}\sqrt{N})$.

afford a direct measure of the improvement effected by a well-designed poppet gear and enlarged steam passages in a simple-expansion locomotive. These curves are therefore also plotted in Fig. 18, which affords a direct comparison with Fig. 14

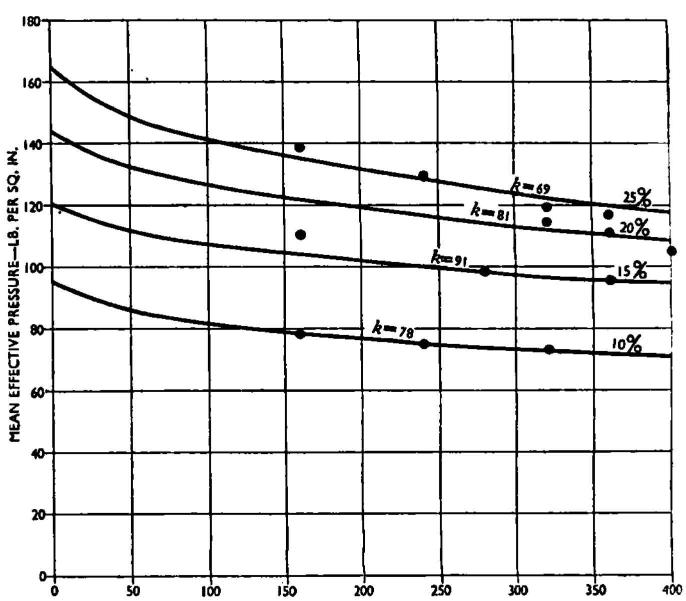


Fig. 19. Mean Effective Pressure Curves for T1 Locomotive with Franklin Poppet Valve Gear

Steam pressure, 300 lb. per sq. in.
Characteristic power equation: $P_N = P_C\,(1-\frac{1}{80}\sqrt{N})$.

showing the results of a similar reconstruction of a compound locomotive.

In 1945 the large four-cylinder simple-expansion express passenger locomotive class T1, referred to in the footnote on

p. 406, was tested at Altoona, and a set of mean effective pressure curves for this engine is given in Fig. 19. This engine was designed from the outset to obtain the maximum power possible at the highest speeds. The Franklin poppet gear was further improved, and the steam passages were designed throughout with this end in view, and without the limitations inseparable from the reconstruction of existing machines. An indication has already been given of the outstanding results achieved, and Fig. 19 enables these results to be compared precisely with the performance of other locomotives, by the method explained in the following section.

A CRITERION FOR CALCULATING POWER AND EVALUATING PERFORMANCE

It has already been mentioned that Dalby's straight-line law for the deviation of mean effective pressure

$$p = c - bv$$

where c and b are constants for a particular engine and cut-off, implies that the mean effective pressure would steadily fall to zero, at a speed of 435 r.p.m. in the case of the locomotive *Schenectady*, and as low as 350 r.p.m. in the case of a locomotive of the Chicago and North-Western Railway for which he also plotted curves. A straight line drawn through the points plotted in Fig. 10 would similarly imply that no indicated power, let alone drawbar power, could be exerted above 400 r.p.m. by the Midland engine.

With superheater engines, at any rate, this is contrary to experience. Curves for mean effective pressure have been plotted for a very diverse and representative range of locomotives in this paper, and the plotted points generally show a rapid drop from the calculated mean effective pressure in the initial stages and tend to straighten out later. Such curves correspond to a simple exponential law, the diminution of mean pressure being proportional to some power of the speed of revolution less than unity :—

$$P_N = P_C - kN^{1/x}$$

where P_N is the mean effective pressure at N r.p.m., P_C is the calculated basic mean effective pressure read from Figs. 1 or 2, and k is a constant characteristic of the particular locomotive. Trial and error showed that $1/x$ does not vary greatly over the whole range of curves, and that the error involved in taking as a mean the square root of N was generally within the scatter of the points.

It was also found that k is a fairly constant proportion of P_C for any particular engine, so that the equation of the curves reduces to the very convenient form

$$P_N = P_C\,(1 - \frac{1}{k}\sqrt{N})$$

The value of k in this equation is shown for every curve plotted, together with the mean value in the form of a "characteristic power equation" for each locomotive, on the diagrams.

This equation not only gives more accurate results over a wide range of speed for superheater locomotives than Dalby's law, but is a more accurate guide in estimating the power at speed of a projected design than either Cole's constants or the Kiesel formula, with its illogical basis in the boiler, provided appropriate values of the constant k are taken. Each designer can establish such values pretty closely from indicator diagrams from his own previous designs, but for general use the following values are suggested :—

Type of engine	k
Simple-expansion with short-lap valve gear, piston or slide valves	25
Simple-expansion with long-lap valve gear, piston valves	45
Simple-expansion with modern design of poppet valves and steam passages of maximum section	75
Compound locomotive with short-lap valve gear, piston or slide valves	30
Compound locomotive with long-lap valve gear, piston valves	50
Compound locomotive with modern design of poppet valves and steam circuit of maximum section	80

The value of P_C may be read directly from the curves in Figs. 1 and 2, but here again the method will give greater accuracy if the value is calculated exactly for the point of compression appropriate to the particular valve gear design adopted. The assumed exhaust pressure must not be modified, however, as this would alter the basis of the constants. The higher the value of k, i.e. the more nearly horizontal the mean pressure curve, and the shorter the cut-off, the more necessary is it that P_C should be accurately determined.

Dalby's law involved two constants, neither of which could be used independently as a criterion of performance. The advantage of a single constant is obvious, but since it has to accommodate two different classes of loss, namely, the pressure drop during steady flow between boiler and valve gear, and the complex cyclic losses between valve gear and atmosphere, any abnormal relationship between the two will be shown up by the fact that no value of k will give a curve conforming with the mean line through the observed values of mean effective pressure. If on the one hand the drop of pressure in the steam chest is relatively small but the steam distribution is bad, the slope of the plotted points will be steeper than the exponential curve given by the characteristic power equation, and will in fact conform more nearly to Dalby's straight lines. On the other hand, if the valve gear gives a good steam distribution, but there is an excessive pressure drop through the superheater and main steam pipes, or excessive clearance, then the exponential power curve will be steeper than the mean slope of the observed mean effective pressures. Examples of both kinds of abnormality are to be found in Figs. 14, 15, and 18, and have already been referred to.

In modern well-designed locomotives such abnormalities generally do not occur, and, as has been seen, the characteristic power equations fit very closely the observed values of mean effective pressure over the normal range of speed. It may therefore be reasonably claimed as an additional advantage of this method of plotting characteristic power curves that it shows up at once such abnormal losses.

To determine the value of k and hence to plot these curves, a value of mean effective pressure P_N at a speed of say $N = 250$ r.p.m. should be adjudged as falling on the average line through the available data for a particular cut-off. (One observed value only can be taken if no others are available, though the scatter of the points in the diagrams in this paper shows that this may give misleading results.) The value of k is then given by

$$k = \frac{P_C \times \sqrt{N}}{P_C - P_N}$$

P_C being read off from Figs. 1 or 2.

It is sometimes found that the value of k for very short cut-offs differs considerably from the average value for other cut-offs. Such abnormal values should be ignored in determining the characteristic power equation (*a*) because no locomotive is designed to give maximum power at the shortest cut-offs and (*b*) because the probable error in evaluating P_C increases with short cut-offs, mainly on account of the increase in the proportion of clearance steam.

The characteristic power equation gives the mean effective pressure, and hence the power at any speed *for a particular cut-off*. The reason why other proposed power formulae have been related to the boiler is to eliminate this condition and to give the maximum horse-power of which the locomotive as a whole is capable, or the power at a specified rate of evaporation.

The author contends that this is fundamentally wrong in principle. The first section of this paper shows how greatly the economy of the engine is affected, provided the steam distribution is good, by the relationship of cut-off and clearance to boiler pressure. The good designer will choose the dimensions of his engine so as to enable it to give the maximum continuous power he requires at an economical cut-off for the steam pressure he is using. The data in this paper will facilitate such determination. The logical procedure is therefore to state the power of the locomotive at that specified cut-off and to design the boiler to supply sufficient steam to feed the engine at that cut-off at the highest contemplated speeds, taking a maximum rate of evaporation appropriate to the design of the boiler, and not a fixed and universal figure as was, for instance, the practice of the German State Railways.

The coefficient k also offers a criterion whereby the test results of a completed locomotive may be assessed. Thus Fig. 20 presents the performances of all the representative locomotives referred to in this paper on an *absolutely comparative basis* by plotting the mean effective pressures as a percentage of the calculated mean effective pressures as given by Figs. 1 or 2. Comparative curves of mean effective pressure on a speed basis

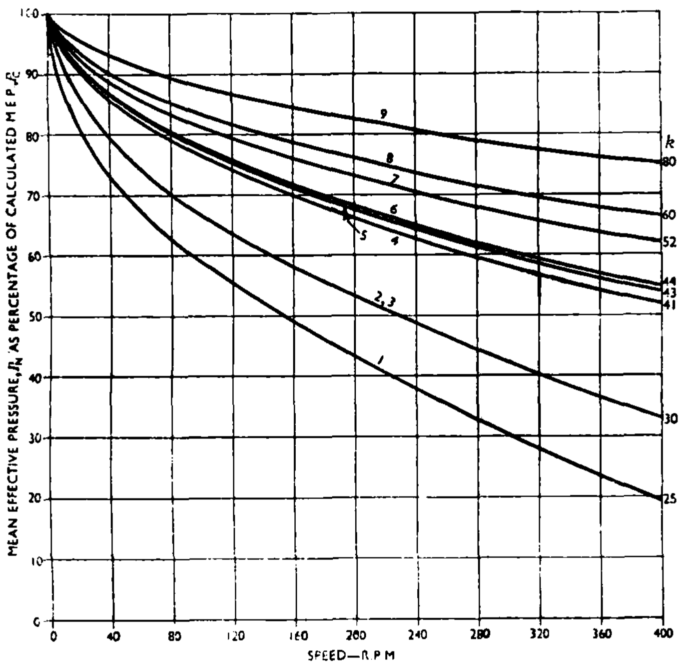


Fig. 20. Mean Effective Pressure Curves for Various Locomotives on Truly Comparative Basis

The values of k in the characteristic power equation for each locomotive are given alongside the corresponding curve.

1 Midland 4-4-0 superheater locomotive (1917).
2 Original P.O. compound superheater Pacific locomotive (1909).
3 Pennsylvania K4S locomotive in original form.
4 Pennsylvania K2S locomotive.
5 Pennsylvania E6S locomotive.
6 L.M.S. rebuilt *Royal Scot* locomotive.
7 Pennsylvania K4S locomotive rebuilt with Franklin poppet valve gear.
8 Rebuilt P.O. compound locomotive No. 3705 (Chapelon).
9 Pennsylvania T1 locomotive with Franklin poppet valve gear.

have frequently been published in recent years as a percentage of boiler pressure. Such curves, however, are not comparable, even if they are for the same nominal cut-off, because, as has been shown in the first part of this paper, the percentage of the boiler pressure attainable as mean effective pressure without loss will vary with the boiler pressure and clearance for the same nominal cut-off.

It is important to emphasize, however, that k is not a direct index of thermal efficiency, but rather of the power-weight ratio at high speed, which for railway operation is considerably more important. Such a coefficient derived from test results should also be related always to the designed characteristics of the locomotive, as these are subject to practical limitations which the designer has to take into account. A high value of k, for instance, is not necessary for shunting or banking engines.

The relationship of the coefficient k to the relative cylinder efficiency is of some interest in connexion with the use of the data curves for steam consumption (Fig. 6) as a basis for boiler design. Comparison with published test results shows that the *minimum* recorded steam consumptions are remarkably close to the figures given by the data curves for optimum conditions, except for early locomotives like the Midland engine of 1917 which have a low value of k. Even under conditions of relative inefficiency, as at late cut-offs, agreement is good if allowance is made for the difference between nominal and effective cut-off. Thus, Chapelon has recorded a best steam consumption of 11·7 lb. per i.h.p.-hr., and all the minimum steam consumptions recorded at Altoona are very close to the data figures. It may therefore be said that provided the valve gear is of modern design and does not give rise to excessive back pressure at speed, the steam consumptions given by Fig. 6 are closely attainable in practice at speed. Nor is this surprising, for the losses of mean effective pressure due to attenuation of the indicator diagram are, in the case of superheated steam, largely made up of throttling losses at admission, and back pressure during exhaust. The first only partially represents a loss of efficiency, as the loss of area is partly compensated by the reduction in the quantity of steam admitted as well as the resulting increased superheat, whereas the second represents a dead loss of efficiency.

Test plant data, however, show that at any particular cut-off the steam consumption may vary by over 25 per cent, due to various causes, particularly varying steam temperature and leakage, and that it generally is higher the lower the speed. Moreover it is well known in practice that the condition of maintenance of a locomotive has an overriding effect on steam consumption, leakage being a principal factor. A margin of at least 30 per cent must therefore be allowed over the minimum consumption given by the data curves.

Acknowledgements. The author is greatly indebted to the following gentlemen who have kindly supplied the data, much of which has not previously been published, on which the second part of the paper is based: Sir William Stanier, F.R.S., Hon. M.I.Mech.E.; the late Mr. C. E. Fairburn, M.A., M.I.Mech.E.; Monsieur André Chapelon, to whose work extended reference is made in the text; Mr. Ralph P. Johnson, Chief Engineer of the Baldwin Locomotive Works; Mr. H. W. Jones, Chief of Motive Power, Pennsylvania Railroad, and Mr. E. W. Marten, Associated Locomotive Equipment, Ltd. The author also wishes to acknowledge the valuable suggestions made by several Members of the Institution who read the paper in manuscript; many of these have been adopted in the published text.

APPENDIX

REFERENCES

ALCOCK, G. W. 1944 Jl. Inst. Locomotive Eng., vol. 34, p. 5, "Development of the Locomotive Poppet Valve Gear in America".

CHAPELON, A. 1936 Mémoires de la Société des Ingénieurs Civils de France, vol. 89, p. 103, "Les Progrès Récents de la Locomotive à Vapeur".

CLARK, D. K. 1852 Proc. I.Mech.E., pp. 60, 109, "Expansive Working of Steam in Locomotives".

COLE, F. J. 1914 American Locomotive Co., Bulletin No. 1017, "Locomotive Ratios". See also "Locomotive Handbook", American Locomotive Co., 1917, p. 54.

DALBY, W. E. 1905 Proc. Inst. C.E., vol. 164, p. 329, "The Economical Working of Locomotives".

DIAMOND, E. L. 1935 *Railway Gazette*, vol. 62, series of articles commencing p. 691, "The Horse-Power of Locomotives—Its Calculation and Measurement".

GOSS, W. F. M. 1902 International Railway Congress, p. 482.

LIPETZ, A. I. 1933 Trans. A.S.M.E., vol. 55, RR-55-2.

NORDMANN, H. 1938 *Organ für die Fortschritte des Eisenbahnwesens*, vol. 93, p. 223.

SANFORD, D. W. 1931 Jl. Inst. Locomotive Eng., vol. 21, p. 217, "Development of the Piston Valve to Improve Steam Distribution".

STANIER, W. A. 1941 Proc. I.Mech.E., vol. 146, p. 50, Presidential Address.

WINDLE, E. 1931 Jl. Inst. Locomotive Eng., vol. 21, p. 378, "Some Notes Relating to Cylinder Performance".

WOOD, A. J. 1925 "Principles of Locomotive Operation", New York, 2nd ed., p. 31.

Discussion

Mr. R. C. Bond, M.I.Mech.E., opening the discussion, said that the paper provided one further proof of the fallacy so often expressed in the past: that steam locomotive design had lain fallow for the last thirty years. There could be no question that as higher pressures were used the relationships between mean effective pressure, cut-off and clearance volume became very important indeed, and the author had clearly shown in detail the relationships which had been known to exist, in a general way, as a result of the tests carried out with dynamometer cars and on stationary testing plants in the past.

In his earlier papers, in which he assessed the losses occurring in the cylinders of steam locomotives, the author had used the Rankine cycle, but on the present occasion he had gone over to something not unlike the locomotive cycle first suggested by Lawford Fry.

While the Rankine cycle could never be entirely replaced as a basic standard of comparison, it was not altogether convenient for a locomotive, which must work at varying degrees of cut-off and usually with some losses due to incomplete expansion. There could be no doubt that cycles such as that of the author or of Lawford Fry had considerable advantages in relation to the investigation which the author had carried out, but he felt that the author's cycle was not a theoretical cycle in the full sense of the term, nor did it represent the best which could be obtained from a given cylinder working under given conditions—because the back pressure assumed was not atmospheric, and the relationship between cut-off and compression had been chosen in a purely arbitrary manner. He felt that the author would have achieved a more satisfactory standard with which to compare the results of his locomotives in service if he had used the method put forward by Da Costa*. Probably the author's calculated steam consumptions (Table 1) would not then have been higher than those of the tests with the German State Railways locomotives. Any standard of comparison ought to be something which one could begin to reach but never exceed.

The author had rightly drawn attention to the importance of clearance, but, had he ensured the most efficient relationship between cut-off and compression, he would probably have found that clearance would not have quite so exaggerated an effect as Figs. 1, 2, and 6 seemed to show. To illustrate this point Figs. 21 and 22 were presented. The values given by the points

* Da Costa, G. 1939 Jl. Inst. Locomotive Eng., vol. 29, p. 399, "The Indicator Diagram and the Efficiency of the Non-Condensing Simple Expansion Steam Locomotive".

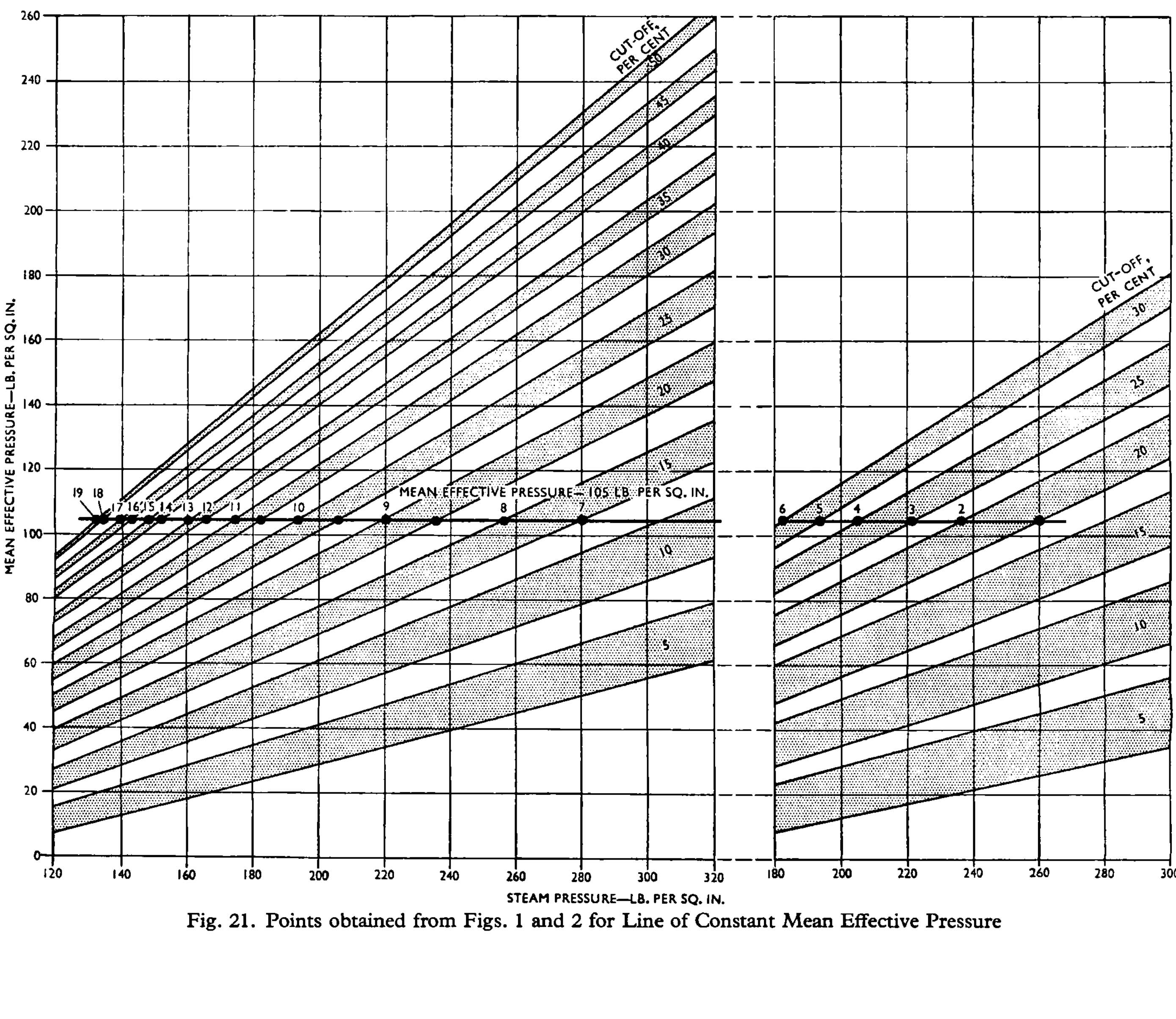


Fig. 21. Points obtained from Figs. 1 and 2 for Line of Constant Mean Effective Pressure

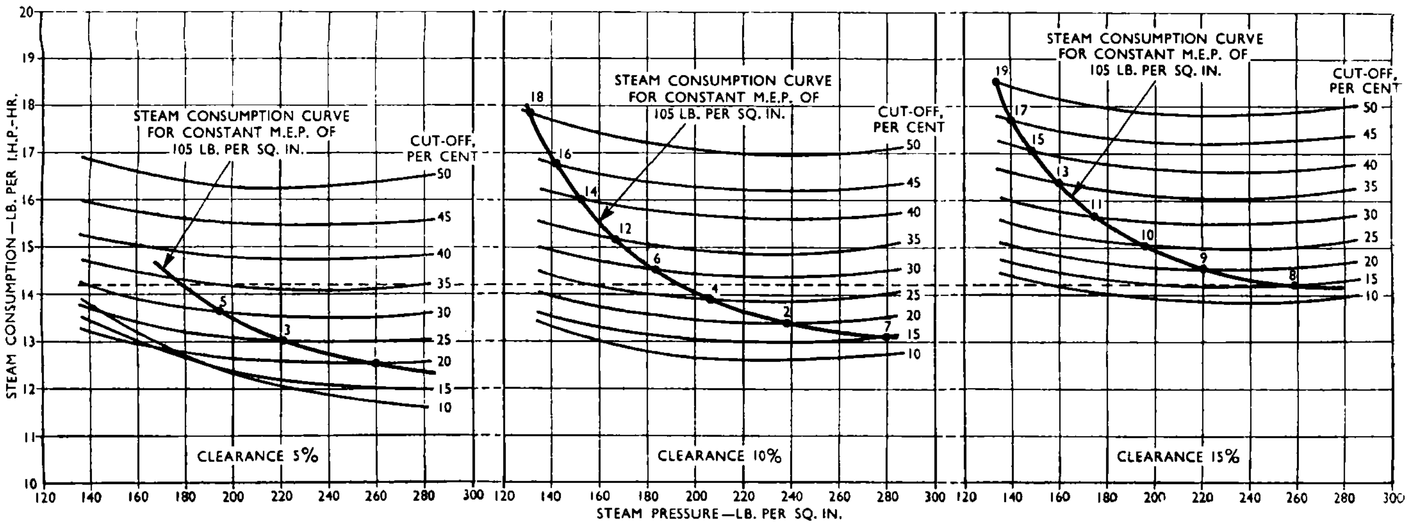


Fig. 22. Specific Steam Consumption for Constant Mean Effective Pressure

at which a horizontal line of constant mean effective pressure crossed the diagonal lines of mean effective pressure in Fig. 21 had been transferred to the steam consumption curves in Fig. 6 of the paper, and were shown in Fig. 22.

The author's diagrams showed that a locomotive at 260 lb. per sq. in. pressure, with 15 per cent cut-off and 15 per cent clearance, actually required more steam than a locomotive at 180 lb. per sq. in. pressure cutting off at 35 per cent with 5 per cent clearance. It did not seem probable that those results would be confirmed in practice generally.

He was very glad to see that the author emphasized the futility of increasing boiler pressures unless the cylinders and valve gear were designed to take full advantage of that increased pressure. He thought that that had been generally recognized in the reduction in the factor of adhesion from about 5 with 20-ton axle loads, twenty years ago, to 3·75 today with 22½-ton axle loads. That went to show that with cylinder capacity as it was today it was possible to take full advantage of the expansive properties of the steam and the available heat drop. To illustrate this point farther, Figs. 23 and 24 were presented. They

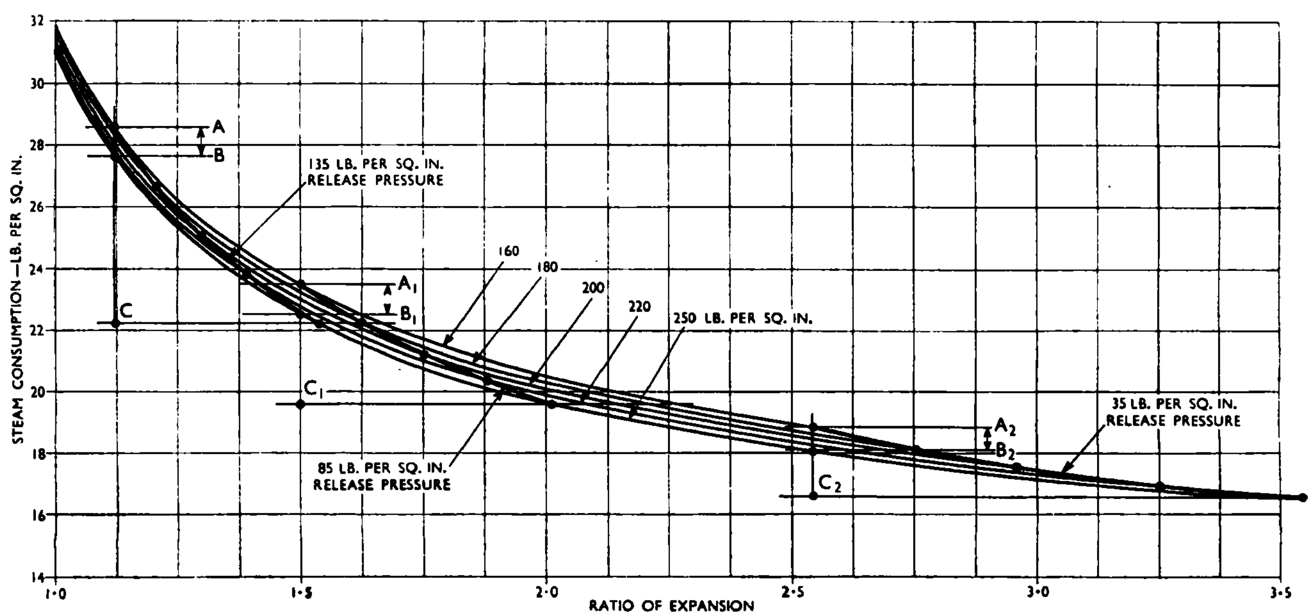


Fig. 23. Steam Consumption and Ratio of Expansion

AB, A_1B_1, and A_2B_2 represent the saving in steam consumption for an increase from 160 to 250 lb. per sq. in., with equal ratios of expansion of 1·12, 1·50, and 2·54, equivalent to release pressures of 135, 85, and 35 lb. per sq. in. At 160 lb. per sq. in. the equivalent release pressures at 250 lb. per sq. in. are respectively 212, 138, and 60 lb. per sq. in. AC, A_1C_1, and A_2C_2 represent the saving in steam consumption by expanding steam, at 250 instead of 160 lb. per sq. in., down to the same release pressures of 135, 85, and 35 lb. per sq. in. respectively. The required ratios of expansion are:—

Release pressure, lb. per sq. in.	Ratio of expansion, at 160 lb. per sq. in.	Ratio of expansion, at 250 lb. per sq. in.
135	1·12	1·54
85	1·50	2·05
35	2·54	3·54

represented curves of steam consumption calculated according to Lawford Fry's locomotive cycle, and showed the differences in consumption with higher pressures at equal ratios of expansion and at equal release pressures respectively.

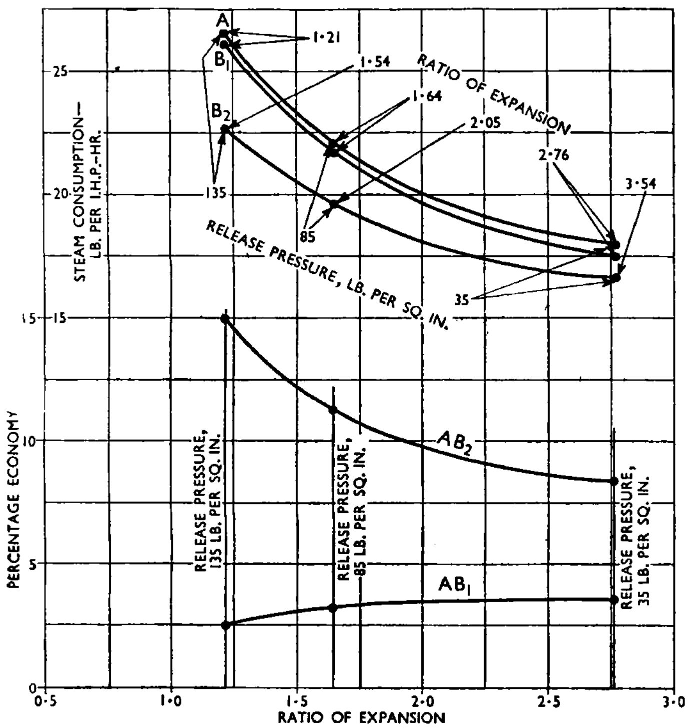


Fig. 24. Percentage Economy of Steam at 250 lb. per sq. in. as compared with Consumption at 180 lb. per sq. in. for Same Release Pressures and Ratios of Expansion

Curve A is for steam consumption at 180 lb. per sq. in. plotted against ratio of expansion, showing three points of release at 135, 85, and 35 lb. per sq. in.
Curve B_1 is for steam consumption at 250 lb. per sq. in. plotted against ratio of expansion, and corresponding with the curve for 180 lb. per sq. in.
Curve B_2 is steam consumption at 250 lb. per sq. in. plotted for the same three release pressures as curve A, and set back on to the same base as curve A.
Curve AB_1 gives percentage economy when working at the same ratios of expansion.
Curve AB_2 gives percentage economy when expanding steam to the same release pressures.

He thought that it was only partially true to say that the use of higher pressures had been brought about only or mainly to ensure the necessary power within the loading gauge restrictions; he would suggest that higher steam pressures had been introduced equally to give the power required at the lowest possible coal consumption and with the greatest possible efficiency.

The progress to which the author had drawn attention was very well shown by the comparison of the indicator diagrams in Fig. 12. It would be seen there how very different the rebuilt Royal Scot locomotive was from the earlier example referred to. It was to be borne in mind, however, that the basic diagram was representative of work done by a very much greater weight of steam than was actually able to force its way into the cylinder. He suggested that the author's diagrams would be improved by the addition of another expansion line, representing the theoretical expansion of that weight of steam which was estimated to be actually in the cylinder.

He would like the author to indicate the steam chest pressure on those diagrams, to show how much of the loss of pressure on admission was due to restriction through the steam ports and how much to restriction through the regulator valve, superheater elements and the steam circuit generally.

The author, in his introduction, recognized the importance of the steaming capacity of the boiler as a final criterion of the ultimate power output of the locomotive. Essentially a locomotive consisted of three power-producing elements: the boiler, the engine, and (the connecting link between the two) the smokebox, chimney and blast-pipe. The efficiency of the last determined very largely the actual performance of the other two; and, while it was most important to utilize the steam efficiently, a boiler that steamed well was even more important. Perhaps that was the reason why in the past, and even to-day, a single curve of mean effective pressure in relation to speed (the limitations to which, at low speeds, were cylinder dimensions and adhesion weight, and, at higher speeds, boiler capacity) remained sufficient for most purposes of design and load determination.

Mr. E. W. Marten (London) said that sustained power at speed in modern locomotive design was not, as formerly, so much a question of boiler capacity as of cylinder performance, the yardstick of which was mean effective pressure, and to maintain this at a maximum figure over the speed range the steam had to flow with the minimum of restriction through passages and ports of adequate proportions, and further, correct relationship and accurate timing of the valve events was imperative.

The author pointed out that his main assumption in this respect related to the point of compression as shown by the inset curve in Fig. 1, which, he stated, was desirable for any type of gear. That, however, was open to question, since from high compression, limiting therefore short cut-off working, the curve fell to zero at something over 60 per cent cut-off.

In actual practice some compression was, of course, essential throughout the range, but otherwise the characteristic of the author's curve well illustrated that, even with a modern Walschaerts gear, it was all a matter of compromise since, with this design, there must be a rapid rise to excessive compression at the shorter cut-offs.

On the other hand, with valve events independently controlled (as with cambox-actuated poppet valves) a closer approach to the ideal was possible, the degree of compression, for a given clearance volume, being made to vary and correspond to the particular cut-off. Since compression aided fuel economy, and as the greater percentage of compression was (as the author remarked) equivalent to a lower percentage clearance, the importance of clearance as stressed in the paper appeared to omit the practical consideration that, in insisting on clearance being an absolute minimum, a loss by wire drawing might be involved where the development of maximum power required large valves. It would be of interest to know how the author would suggest overcoming the difficulty of achieving in cylinder design what in effect he advocated, namely maximum steam-flow area and, at the same time, minimum clearance volume—two conflicting requirements that called for a compromise—for the lowest clearance volume consistent with the large valve and passage areas.

The paper emphasized that the exact point of compression was all-important to the accuracy of the results to be determined by the author's method, and although the foregoing remarks affected somewhat the value of the coefficient k, his equation curves appeared generally to agree well with the selected tests, and the same might be said in regard to some test records in the writer's experience. But the more modern designs cited were confined to locomotives having piston valves or poppet valves operated by Walschaerts gear, or with gear working on a similar principle. To these examples might be added cambox-operated poppet valves working on the rotary-cam principle, the latest application of which was soon to be seen on main-line locomotives in this country. Compared with the areas of Fig. 13, the noticeable increase possible with this system, in the maintenance of port openings down to the shortest cut-offs, was apparent from Fig. 25. Coupled with the fact that the lead also increased on notching up, adequate head of steam was therefore assured.

The best showing in the paper was with the poppet valve; this demonstrated again what was invariably found in comparative trials: that for any given cut-off and speed the poppet-valve gear produced a higher mean effective pressure, and hence a higher horse-power in the cylinders, than did Walschaerts valve gear.

The results with the T.1 class locomotive in particular served as a classic example of improvement in power output and economy as a result of design of cylinders and valve gear, with

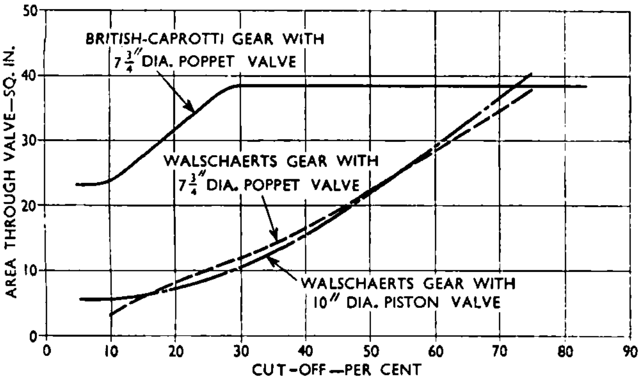


Fig. 25. Comparative Inlet Valve Areas

correct proportioning of areas throughout the circuit from regulator head to blast-pipe. This design gave the lowest water rate consumption in the forty years of test experience at Altoona.

Mr. E. S. Cox, M.I.Mech.E., referred to the rebuilt Royal Scot class of 4-6-0 engines on the L.M.S. In modernizing the original Royal Scot engine (itself a notable performer in its day) it was preferred to improve the performance without sacrifice of economy, rather than to aim at still higher efficiency without improvement in performance. As a result, the operating department had a locomotive whose mastery of its work, consistent performance, and reserve of power, made it a much more valuable traffic tool than its predecessor, and so raised the overall efficiency of passenger train operation; this was just a reminder that it was necessary to consider efficiency in somewhat wider terms than those outlined by the author.

The increases in steam areas, as compared with the original engines, were:—

Main steam pipe in boiler	36 per cent
Superheater elements	45 „
Steam pipes in smokebox	49 „
Steam ports in valve liners	34 „

Valve gear dimensions were as followed:—

Valve gear dimensions, inches	Original	Rebuilt
Steam lap	$1\frac{7}{16}$	$1\frac{3}{8}$
Lead	$\frac{1}{4}$	$\frac{5}{16}$
Exhaust clearance . . .	Line and line	$\frac{1}{16}$
Maximum travel	$6\frac{3}{8}$	$6\frac{3}{8}$

It would be seen that a slight retraction had been made from the usual preoccupation with getting the steam out of the cylinders, and some attention had been paid to getting more in, a point to which Chapelon appeared to have given attention in his latest designs.

On a series of dynamometer-car tests carried out in 1945 with 450-ton trains between Crewe and Carlisle, the rebuilt engine showed a modest coal economy of 5½ per cent over the original type. The tests, however, only covered rostered loadings on existing timings well within the capacity of the engines; no opportunity had arisen to carry out any full-power tests with this class.

The author indicated the great increase in power output per unit of cylinder volume which had taken place in recent years, and while the outward aspect of locomotives had not greatly changed in the past twenty-five years, it was this continuous and large improvement in the power/weight ratio which had kept the steam locomotive in use.

Fig. 26 illustrated this point in a slightly different manner, using the well-known type of mean effective pressure/speed curve which took into account boiler, as well as engine, power.

Curve 1 was still being used for train loading calculations in the drawing office of the railway on which he had served his time, and the superheated L. and Y. engines of that day, having Joy's gear with 1-inch lap, restricted ports and passages, large cylinders, and small boilers could not do much more.

Curve 3, based upon numerous series of dynamometer-car tests, showed what a modern narrow firebox locomotive with

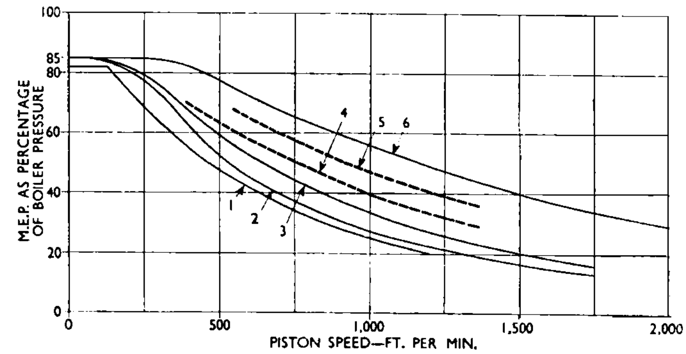


Fig. 26. Comparison of Various Mean Effective Pressure Curves

Mean effective pressure as percentage of boiler pressure at 1,000 ft. per min.:—

1. Old Lancashire and Yorkshire Railway curve (pre-1914) . 25
2. L.M.S. standard curve for pre-1927 locomotives . . 27·5
3. L.M.S. standard curve for modern locomotives . . 33·5
4. Values obtained from actual tests with 5XP locomotives, L.M.S. . . . . 39·5
5. Values obtained from actual tests with 7P locomotives, L.M.S. . . . . 47·5
6. R. P. Johnson's curve based on test plant data representative of modern American locomotives . . . 56

long-lap valve gear could sustain in daily service, and was the one used on the L.M.S. in train loading calculations for its most modern engines.

It was very rarely in this country that locomotives were operated at full power for any length of time—time-tables were always arranged to leave a considerable margin between what the locomotive was called upon to do and what it could achieve when all out—and this seemed common sense if reliability and low maintenance costs were to be ensured.

Two series of special tests had, however, been run on the L.M.S. at full power output with a Class 5X engine in 1937 and with a Class 7 engine in 1939, and curves 4 and 5 were derived from these tests. The upper one represented about the limit which could be sustained with hand firing. Curve 6 was taken from R. P. Johnson's book "The Steam Locomotive", and showed representative modern American practice, where the loading gauge and allowable weight greatly increased the boiler size possible, and where the universal use of mechanical stoking on large engines removed all human limitations to the firing rate.

At 1,000 ft. per min. piston speed, the percentage of boiler pressure available as mean effective pressure for the L.M.S. Class 7 4-6-2 was just about twice that of the old L.Y.R. engines, thus confirming the author's figures.

In Fig. 27 he had superimposed the two curves 4 and 5 of Fig. 26 on to the author's Fig. 11 giving mean effective pressure curves for the rebuilt Royal Scot locomotive. He had added two further curves for 40 and 50 per cent cut-off according to the author's calculations, and the author's curves, while they might measure cylinder efficiency, formed no measure of locomotive power output. In order to obtain the latter it was necessary to know up to what speed each different cut-off could be sustained without drop in boiler pressure; clearly it was not possible to proceed very far with locomotive power calculations without considering the boiler.

Allowing for boiler capacity over a range of speed from 35 to 72 m.p.h. in the full power tests of the Class 7 engine already referred to, it was clear that an area something like ABCDE was necessary to define the power range of this locomotive. Such a diagram as this could only be accurately drawn by means of constant-speed testing, but the area indicated was probably farly representative for this particular engine.

Now, although the valve gears and ports and passages were not identical in the Class 7, four-cylinder and the Class 6, three-cylinder rebuilt Scot engines, the steam consumption over a series of variable speed tests was within 2 per cent for the two classes so that the mean effective pressure for a given cut-off was probably very similar for the two engines. The cut-offs shown indicated that, in fact, the curves for modern simple-expansion piston valve engines could lie somewhat higher than the author suggested and that the representative factor k in the author's table was too low.

This was the more understandable when it was realized that the lines shown on the author's Fig. 11 were not based on any full-power tests, but were built up from a series of indicator

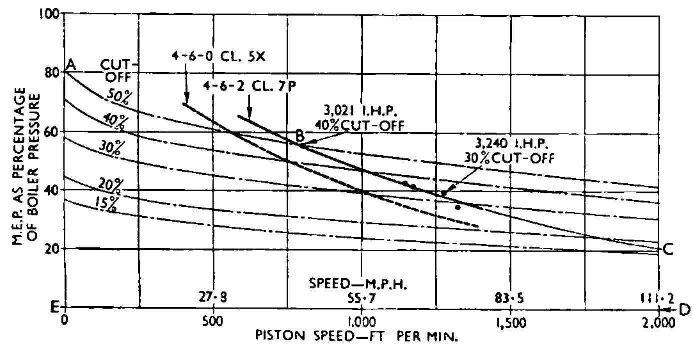


Fig. 27. Actual Performance of L.M.S. Classes 5X and 7 in Relation to Fig. 11

cards taken fortuitously at certain speeds and not related to the maximum speed at which the boiler was able to sustain any particular cut-off. The three points which he wished to make were therefore:—

(1) That it was essential to consider the boiler in any consideration of locomotive power at speed.

(2) That the modern simple-expansion, piston-valve locomotive was able to sustain a higher relative position than that allotted to it in Fig. 20.

(3) That in studying cylinder performance only, by means of such curves as Fig. 11, cards were required at each cut-off up to the speed at which the boiler would no longer supply the engine, in order to establish the correct shape of the curves in the upper speed range.

The whole of this kind of work would be brought out of the range of opinion, and could be based on first-hand accurate information when the Rugby testing station became available, but the author had suggested a very interesting method of making comparisons on cylinder performance alone.

Mr. E. C. Poulteney, O.B.E. (London), said that it was necessary to take into consideration the boiler capacity. If the curve of the mean effective pressure were plotted against speed, unless the boiler capacity were known it would not be possible to sketch the ultimate tractive effort lines, because one did not know, in the absence of steam-supply data, what speed would be obtained for a certain cut-off. With a very large boiler it was possible to run at greater speed at a longer cut-off corresponding to the higher indicated horse-power.

The rest of his remarks he proposed to devote to some practical points. The valve liner used on the New York Central System for their newest locomotives had sharp corners on the steam admission edges (Fig. 28). It was for a 25-inch cylinder; the area through the bush was 90 sq. in., and through the port 53 sq. in., and he thought that that showed a very good ratio, because naturally it was desirable to have as much steam area through the bush as possible at short cut-offs, to make the best use of the cylinder port area.

Another point of importance was steam chest volume, and he pointed out that the steam chest of the Castle class 4-6-0 locomotives of the Great Western Railway was bulged out to give extra steam volume (Fig. 29). It must always be remembered

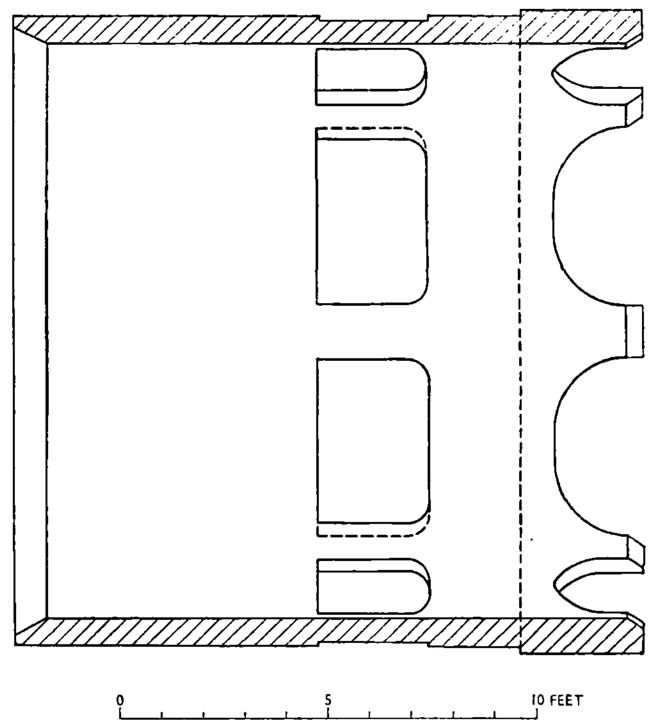


Fig. 28. Valve Liner used on New York Central 4–8–4 Locomotive

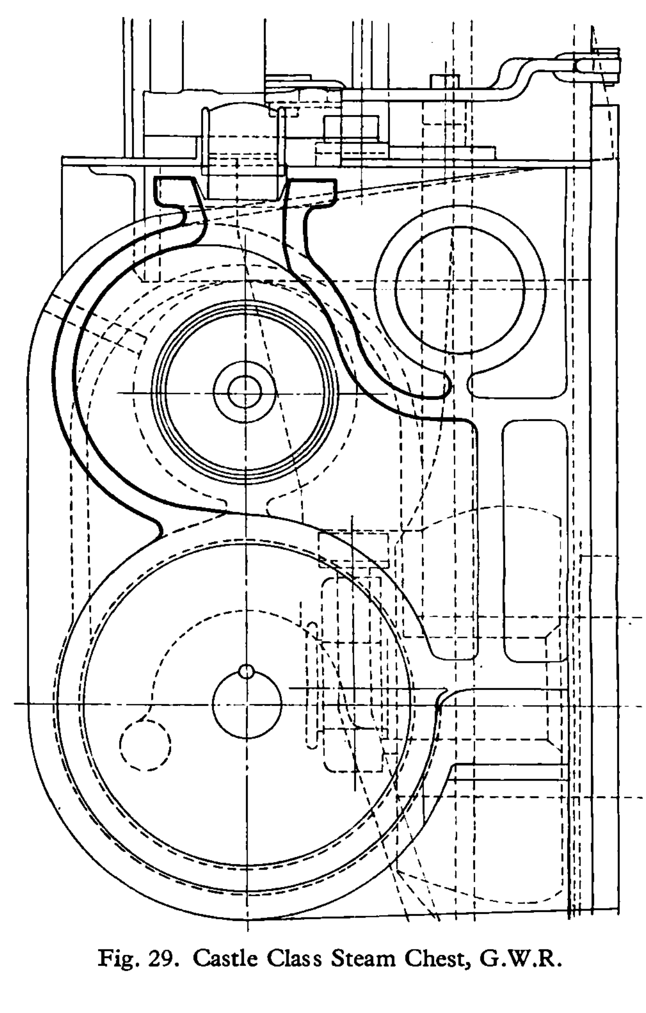


Fig. 29. Castle Class Steam Chest, G.W.R.

that it was the steam in the steam chest which went into the cylinder, and not the steam in the pipe.

In the Pennsylvania Railroad's E6s Atlantic, the steam chest volume had been added to by bulging out the casting in order to provide room for as much steam as possible (Fig. 30).

The Pennsylvania T1 4-4-4-4 locomotive, to which the author had referred, developed a maximum horse-power of

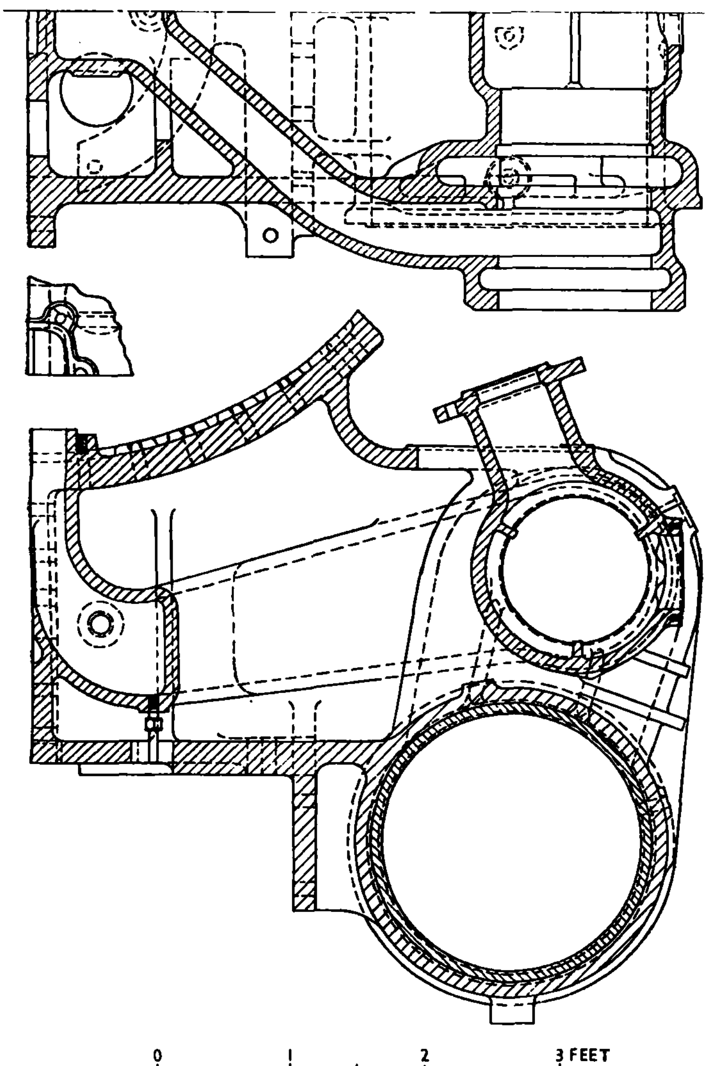


Fig. 30. Pennsylvania Railroad E6s Atlantic Cylinder

6,552, at 86 m.p.h. and 25 per cent cut-off. That might not sound very much having regard to the size of the locomotive, which was very large, but it should be borne in mind that the cylinders were of only 19¾ inches diameter and 26 inches stroke, and each one of those cylinders indicated 1,630 h.p., so that if one had an ordinary two-cylinder engine with that valve gear, and with cylinders no larger than 19¾ inches × 26 inches one could get 3,276 h.p. out of it, and that did mean something. To attain this high power, however, it would be necessary to have a steam supply of just over 50,000 lb. per hour; this showed how important was the boiler.

Professor G. V. LOMONOSSOFF, Dr. Ing., M.I.Mech.E., said that the author's condensed but comprehensive paper covered four subjects, each of which could be developed into a volume. The first was the question of the ideal cycle for the locomotive. In this country the Rankine cycle was still in common use, although it was not realizable in reciprocating engines. In that respect Zeuner's cycle was better, but it ignored the influence of the clearance and compression. Twenty years ago Lawford Fry proposed his "locomotive cycle", but on that occasion Mr. Diamond justly pointed out that it was too elastic and indefinite*. The author now proposed his own cycle, which he himself thought was a more practicable one. He hoped that it would remain in common use even longer than that of Rankine.

The second and most important subject was the recent improvement of locomotive steam distribution. There again he was in full agreement with the author, but he thought that it would be advisable to mention the name of M. Ricour, C.M.E. of the French State Railway, who in 1886 showed in which direction that development should go†.

The third subject—to which most of the author's diagrams referred—was the phenomenon of distortion of indicator diagrams due to turbulence in the steam flow, which increased with the speed. That fact, however, was established not by Goss and Dalby but by D. Clark in 1855‡, and was carefully examined in 1890 by Desdouits§, in 1897 by Barbier||, and later by many other investigators, including Nadal¶ and himself**. In accordance with all those tests, except those of Desdouits and Doyen, the tractive effort F fell, when the speed increased, according to the parabolic law $F=F_0-CV+CV^2$ (curve P in

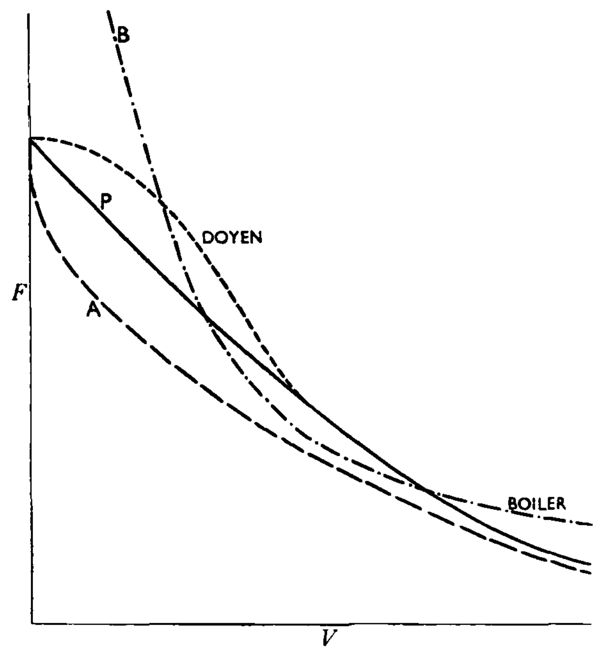


Fig. 31. Relation of Tractive Effort and Speed

Fig. 31, where for wet steam C is very small). But he agreed that the author's law, $F=F_0-\frac{\sqrt{V}}{k}$ (curve A) was not only simpler, but at speeds over 120 r.p.m. agreed with the results of his own tests better than parabola P. As to low speeds, the Doyen curve seemed the most probable††. For instance, it satisfied the points of Fig. 10 much better than the author's curve.

In order to be comparable, each of these relations should correspond to the same conditions. As he understood it, however, some of the author's curves related to variable openings of the regulator. That greatly diminished their practical significance; according to experiments by Barbier and himself, the

* DIAMOND, E. L. 1927 Proc. I.Mech.E., p. 510.

† RICOUR 1886 *Revue Générale des Chemins de Fer*, vol. 9, part 2, p. 212.

‡ CLARK, D. K. 1855 *Railway Machinery*, p. 69.

§ DESDOUITS 1890 *Revue Générale des Chemins de Fer*, vol. 13, part 1, p. 275.

|| BARBIER, F. 1898 *Revue Générale des Chemins de Fer*, vol. 21, part 1, p. 35.

¶ NADAL, J. 1903 *Revue Générale des Chemins de Fer*, vol. 26, part 1, p. 301; 1904 *Revue Générale des Chemins de Fer*, vol. 27, part 1, pp. 180, 208; part 2, pp. 165, 183.

** LOMONOSSOFF, G. V. 1926 "Lokomotioversuche in Russland", pp. 84 and 153.

†† BLUM, VON BORRIES, BARKHAUSEN 1903 *Die Eisenbahntechnik de Gegenwart*, p. 347.

opening of the regulator considerably changed the values of b, c, and k*, and the full opening was not always the optimum one.

The fourth subject of the paper was the universal power formulae (locomotive ratios) such as those of Frank, Cole, Kiesel, and Lipetz. Personally, he had fought against them since 1903, but he thought that the author's criticism was too severe, and not always fair. The author did not mention that there were countries which did not use such formulae in principle—Russia, Poland, France, Italy, and, since 1923, Germany. In all those countries the railway authorities preferred to test each new class of locomotive scientifically, and to use the data thus obtained to improve the class and to design new ones.

No one had ever used those formulae for designing cylinders and steam distribution. Even when, before 1923, the formulae of Frank and the tables of von Borries were in common use in Germany, Bruckmann's curves were recommended* for this purpose. The data of Frank and von Borries were used only for the determination of the heating surface. Thirdly, there was a domain where universal formulae were almost unavoidable; this was in the designing of new railways, especially in the Colonies, when often nobody knew what types of locomotives would work on them.

Fourthly—and this was most important—the curves representing such universal formulae (locomotive ratios) or corresponding boiler curves, obtained experimentally (B in Fig. 31), could in no case be compared with the engine curves A, P, D, to which the paper was mainly dedicated. Each engine curve should correspond to constant position of regulator and gear but to varying rates of evaporation. On the other hand, each boiler curve corresponded to different cut-offs and openings of the regulator, but to fixed boiler conditions. Consequently the boiler curves never crossed the axis F (Fig. 31)†, whereas all engine curves unavoidably did so.

It was interesting to note that the Kiesel formula applied to Russian locomotives almost coincided with the maximum boiler curves obtained experimentally with the oil firing.

It was necessary to be very careful about the use of the term "mean effective pressure". The author appeared to apply the term to the mean indicated pressure p_i, obtained by planimetration of the indicator diagrams; but on the Continent and in America it was understood in the sense that $p_e=\eta p_i$, where η, which varied between 0·85 and 0·97, was the mechanical efficiency between cylinders and the rail.

The author's remark in the second paragraph of the last page of the paper was very important. Of course, coefficient k characterized only the perfection of the steam distribution and concerned neither internal friction nor the missing quantity. Personally, however, he could not agree with the author's words "leakage being a principal factor". When, in the Esslingen laboratory in 1924, leakage was reduced to zero, the missing quantity, especially at early cut-offs, was very far from zero. With high superheating, of course, there was no condensation, but contraction of steam took place, and that contraction, like condensation, rose with the compression. This latter fact was established by C. A. Borodino‡ and confirmed by special experiments by Professor Dwelshouvers Dery§.

Dr. F. C. Johansen, M.I.Mech.E., said that the author's methods of presenting the effects of clearance, steam pressure, and speed on mean effective pressure were noteworthy.

With regard to the first of those factors, exemplified by Figs. 1 to 4, it was an instructive exercise to consider how the physical agents involved led to the conclusion that the effect of a small change in clearance volume was greater for small than for large percentage clearance space, greater for early than for late cut-off, and greater for high than for low steam pressure. The trends of design towards small clearance, early cut-off and higher steam pressure therefore enhanced the critical effects of clearance and pointed to the care with which it needed to be considered in future work.

* Lomonossoff, G. V. 1926 "Lokomotioversuche in Russland", p. 170 and pp. 122, 165.

† Lomonossoff, G. V. 1926 "Lokomotioversuche in Russland", p. 114.

‡ Borodino, C. A. 1886 Proc. I.Mech.E., pp. 318, 349.

§ Dery, D. 1897 *Revue de Méchanique*, p. 925; and 1900 *Revue de Méchanique*, p. 5.

With regard to the general effect of speed on mean effective pressure, he would like to hear some discussion of the author's basic formula. The k had the same dimensions as $\sqrt{N}$ (namely $T^{-\frac{1}{2}}$) and was therefore hardly to be described as a coefficient, but the author said that k was an index of power/weight ratio, having the dimensions of speed. However, k, or some equivalent quantity, was in fact an index of power/weight; it should have the dimensions of speed, so that the formula could be made rational by reconstructing it with piston speed in place of $\sqrt{N}$.

Power of an engine did not fall off at high speed primarily as a result of the frequency at which the steam entered and left the cylinder, but because it failed to enter the cylinder fast enough to keep pace with the rapidly moving piston without fall in pressure. The high-speed indicator diagrams of Fig. 12 seemed to support the view that the steam in the cylinder could neither attain its theoretical maximum pressure at any stage of admission, nor maintain its actual maximum pressure throughout admission. The rational base against which to plot the diminution of mean effective pressure was probably mean piston speed, as used by Dalby (Fig. 9) rather than r.p.m., as adopted by the author. More logical perhaps, on physical grounds, was the speed of flow of steam, which depended on the rate of change of cylinder volume as the piston moved.

It was clear that, for an engine of any given dimensions and geometry, r.p.m. was a measure of both piston speed and rate of change of cylinder volume. The author, however, allocated representative values of k to various *types* of engine, and much of the practical utility of his formula resided in its use for comparing performances of different locomotives of the same type and for inferring the power-speed characteristics of projected locomotives from the known characteristics of existing ones having the same sort of expansion and valve gear. Within a single type, so defined, considerable variations in size and proportions were practicable, and the power-speed characteristic would, in general, differ according to the speed which was used as a base.

The point might be illustrated by comparing three classes of locomotive—the G.W.R. Hall class, the L.M.S. Royal Scot, and the Southern Railway Merchant Navy—all of which belonged to the author's "simple expansion with long lap valve gear and piston valves" class. Among those three locomotives, for equal r.p.m., the greatest disparity amounted to 12 per cent in respect of speed of travel along track, 25 per cent in respect of average piston speed, and 32 per cent in respect of average rate of change of swept cylinder volume. The greatest disparity in each case was between the Great Western engine and one of the others; but that did not, apparently, arise from any peculiarity of Great Western design, since if the G.W.R. King class were included in the comparison, it became apparent that a good deal of the disparity was to be associated with the fact that the Hall class engines had only two cylinders, although the disparities among the other engines, having three or four cylinders, were still significant.

On purely physical grounds, the number of cylinders in which power was being developed at high speed seemed to be a factor of sufficient importance to warrant account being taken of it in the assignment of values of k to different engine types, whether the basis of comparison was r.p.m. or some other measure of speed.

Mr. H. Holcroft (Chipstead) remarked that the subject of the paper was but one aspect of the many-sided problems of the locomotive. The parts were so interrelated that to consider one without the others overlooked this interdependence. For instance, improved boiler and superheater design and more efficient smokebox arrangements would result in the easier passage of gases, and therefore permit of a reduced velocity at the exhaust tip; and that meant that there was lower back pressure in the cylinders, and therefore the cylinders had greater capacity and efficiency. He felt that some of the improved results which the author attributed to valves and valve gear might, in part, be due to boiler design.

That was one point which the author had not taken into account, as he had assumed a back pressure of 18 lb. abs. in all his calculations, and that was notwithstanding the fact that in American practice the back pressures were very much higher than were reached in British practice.

The adoption of 250 to 400 r.p.m. in itself raised many problems. For instance, in increasing the revolutions from 200 to 400 r.p.m., the hammer-blow in two-cylinder engines was increased fourfold.

With regard to the author's subject in itself, he had prepared two diagrams. Fig. 32 showed i.h.p. plotted against r.p.m., the dotted line representing a typical locomotive in which the maximum i.h.p. occurred at about 220 r.p.m. The full line related to a small steam rail-car engine with piston valves and link motion, and which was in fact a miniature locomotive itself. The maximum in that case was not reached until about 1,000 r.p.m., notwithstanding the fact that the port opening to steam and exhaust occurred in only one-fifth of the time taken in the locomotive. If those curves were replotted on a basis of mean piston speed in ft. per min., the two peaks were brought very much closer together and came within the 900 to 1,000 range.

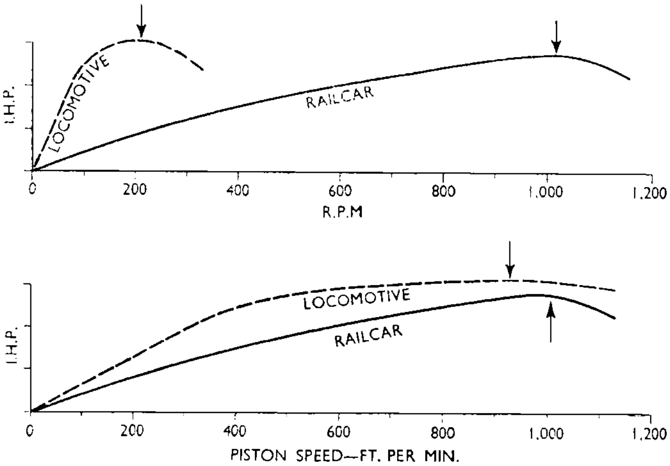


Fig. 32. The Relationship of Indicated Horse-power to Piston Speed and Rotational Speed

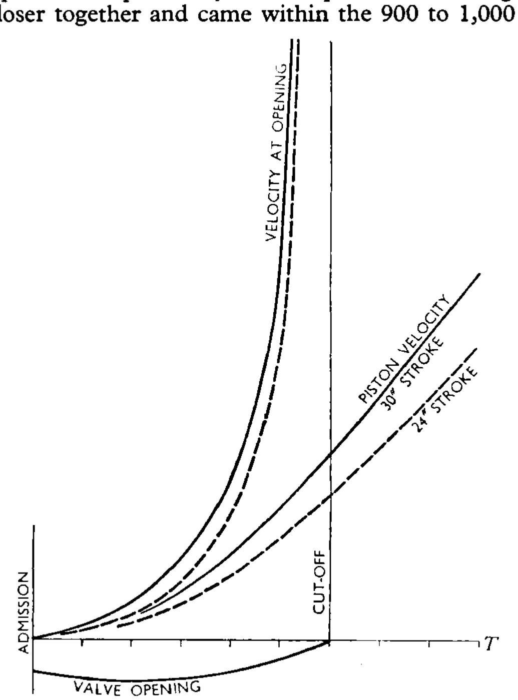


Fig. 33. Velocity of Steam at Port Opening

The point he was trying to bring out was that r.p.m. was not a safe basis on which to plot curves of m.e.p. and i.h.p., and that it was much sounder to take the piston speed instead.

Steam flowed from one region to another only by reason of difference in pressure, and that difference in pressure was created in a cylinder by the moving piston. If the compression space were full at the end of the stroke, the steam entered only as the piston made room for it up to the point of cut-off. He had therefore prepared another diagram, Fig. 33, in which piston velocity and the amount of valve opening were plotted on a time base from admission up to a cut-off of 25 per cent. The circumference of the valve and the area of the piston being constant, the two variables were the velocity of the piston and the amount of valve opening. By dividing the velocity of the piston by the opening there was obtained a nominal curve of the velocity through the opening, if the specific volume of the entering fluid were regarded as constant. It would be seen that the rapidly rising curve went up to infinity when the valve closed, so that, of the right-hand part of the valve opening, the last 15 per cent or so had very little value and should be disregarded. However large the valve or rapid its closing, there was bound to be this rounding off of the corner of the indicator diagram at cut-off. By producing a number of these velocity curves for different piston velocities and cut-offs, it would be possible to assess their relative values by estimating the mean velocity through the opening in each case.

If the full line in Fig. 33 were considered to represent a 30-inch stroke as used in G.W.R. two-cylinder engines, the dotted line represented to scale a 24-inch stroke as used on the S.R. Pacifics. It would be seen that the velocity curve through the opening, with a 24-inch stroke, was very appreciably lower, which meant that the steam got into the cylinder more easily, so that there was a larger quantity per stroke. He thought, therefore, that in some way or other the author should include the factor of stroke in his calculations. It brought in piston speed, which determined in some measure how much steam could get into the cylinder. The author's adoption of revolutions as the basis of his curves held good to the extent that any variation of piston stroke from the average was covered by his general coefficient k, which absorbed all the idiosyncrasies of any particular locomotive, including stroke and back pressure, and that was quite safe in developing a formula for mean effective pressure at various cut-offs in the locomotive in question; but to apply the same coefficient to new design seemed rather a risk.

Dr. W. A. TUPLIN, M.I.Mech.E., congratulated the author on presenting an informative and neatly compressed paper which provoked thought. Among the numerous diagrams given in the paper, Fig. 6 was of particular interest to himself. From this it would be seen that while the steam pressure varied from 140 to 280 lb. per sq. in., the rate of steam consumption varied very little indeed. Actually it varied proportionately even less than that diagram suggested, because the origin was not on the diagram at all, but ten units below. The variation of steam consumption with steam pressure was therefore very low indeed, so that there was not much gain in efficiency by raising the pressure. The diagram was based on a constant steam temperature of 600 deg. F., so that the coal rate was proportional to the steam rate.

Since the efficiency was not appreciably improved by raising the pressure, he had wondered why there had been a tendency to raise it. True the cylinders could be reduced in size, but that seemed to him to be a very small advantage to set against the greater weight of the boiler, which meant that, unless the higher pressure produced higher efficiency (which was not proved) the power/weight ratio of the locomotive went down as the pressure went up. Possibly locomotive engineers had been influenced by Churchward's practice, a feature of which was a rather high pressure, and that had been assumed to be a vital factor; he personally thought that was not the case.

Another consideration which might have influenced locomotive engineers was the fact that in power station practice, over the last twenty-five years, boiler pressures had gone up from 250 to something like 1,000 lb. per sq. in. with very substantial gains in efficiency. That argument, however, was quite fallacious, because the efficiency which was obtained depended almost entirely on expansion ratio, and in a power station it was possible to attain almost any expansion ratio desired by putting in sufficient stages. The locomotive, on the other hand, did not expand down to 1 lb. per sq. in., or even down to atmospheric pressure, but down to a certain fraction of its boiler pressure. It was not possible appreciably to alter the expansion ratio by altering the pressure, so that the efficiency was not, theoretically, improved appreciably when the pressure was raised.

One could usually get a coal consumption of about 3 lb. per drawbar h.p.-hr., and the scatter round that figure seemed to be almost independent of pressure. The designer might present the results in such a way as to conceal unfortunate aspects of the work. Table 1 was a comparison of what happened at 206 lb. per sq. in. and 294 lb. per sq. in. In it the higher steam pressure gave the lower steam consumption throughout the range. At 206 lb. per sq. in., the steam consumption decreased as the cut-off was advanced (the expansion ratio was increased)—in accordance with theory. At approximately constant cut-offs, however, the lower pressure gave higher efficiency for a given expansion ratio.

This was well known to some people*. A change of pressure from 180 to 220 lb. per sq. in. on the L.N.E.R. Pacifics did not, on test, produce any reduction in coal consumption. It might be said that enough had been heard about coal economy lately; that the real criterion of ability to develop power at speed was the ratio of mean effective pressure to steam chest pressure, and that high pressure should improve it. He himself did not think so. The steam had fleeting opportunities of getting into the cylinder through a very narrow opening, and its capacity to get in depended on its quickness "off the mark". He suggested that that would be associated with its energy per unit weight. At a constant temperature of 600 deg. F., the energy in a pound of steam was slightly less at 10 lb. per sq. in. than at 300 lb. per sq. in.; so that high-pressure steam did not move any faster, and was not quicker at getting in. He thought the velocity of propagation of a wave through steam, proportional to the square root of the absolute temperature and independent of pressure, might be a criterion. It was possible to get the steam to move faster by raising the superheat, but the advantage was small—less than 5 per cent if the temperature were raised from 650 deg. F. to 750 deg. F.

If a boiler pressed to 300 lb. per sq. in. were fitted to an engine designed for 200 lb. per sq. in. the performance, as regards acceleration and speed, would improve, provided it did not slip, and the coal consumption might be reduced—the tractive effort had increased; it was possible to work at a lower cut-off, and therefore the efficiency was higher. But that result could be obtained without increasing the boiler pressure, simply by enlarging the cylinders and valves suitably.

He considered a cylinder 20 by 26 inches, with 10-inch valve, $1\frac{1}{2}$-inch lap, and $\frac{1}{4}$-inch lead. At 24 per cent cut-off and 420 r.p.m., for steam to get into the cylinder and fill it to the steam-chest pressure would involve a mean velocity through the port of 1,650 ft. per sec., approaching the speed of propagation of a wave through steam. The steam would not do it. There was no need to bother to take indicator diagrams to find whether the top of the diagram was flat; it could not be flat. Incidentally, the speed with which the steam had to get out of the cylinder was only 300 ft. per sec., which tended to disprove the legend that it was easier to get steam into a cylinder than to get it out.

To obtain a high ratio of mean effective pressure to steam chest pressure it was necessary to have a high ratio of area of port opening to cylinder volume. Secondly, to save coal it was necessary to have a high expansion ratio and no leakage. Thirdly, to save water it was necessary to have a high superheat and no leakage. Fourthly, to secure a high power/weight ratio one needed the lowest practicable boiler pressure and no leakage. High pressures tended to promote leakage.

Mr. O. S. Nock, B.Sc. (Eng.), M.I.Mech.E., referred to one factor which was not mentioned in the paper, namely superheat temperature. He imagined that an engine in which the degree of superheat was lower than in the case of, for example, the L.M.S. converted Royal Scot would not show such good figures, even though the cylinder dimensions were the same and the cylinder proportions generally—ports and so on—were just as good. It would be interesting to have the author's views on that point.

He had in front of him some indicator diagrams taken in 1913 off one of the L.N.W.R. four-cylinder 4-6-0 engines of the Claughton class, which were not generally considered to be very good locomotives. At 58 m.p.h., that locomotive, with 175 lb. per sq. in. boiler pressure, showed a mean effective pressure of 64 lb. per sq. in., which, on the basis of the author's analysis, gave about 50 per cent of the ideal. Against this the Midland 4-4-0 quoted in the paper as a typical example of a pre-1914 locomotive gave only about 35 per cent at the same speed and cut-off. The cylinder performance seemed to have been good on the L.N.W.R. engine, yet the whole class failed, simply because their boilers were not big enough and efficient enough to supply the steam required. The locomotive in question had four cylinders 16 inches in diameter and 26 inches in stroke, maximum valve travel $4\frac{5}{16}$ inches, lead $\frac{5}{32}$ inch, steam lap 1 inch, and exhaust clearance $\frac{1}{16}$ inch, while the indicated horse-power at 58 m.p.h. was 1,606. The above results rather suggested that in any method of predicting performance the whole engine had to be considered, rather than one portion of it; it also went to show how important it was, in analysing locomotive performance, to take each portion of the machine—the boiler, firebox, front end, and so on—by itself, rather than treating the whole thing on the basis of the drawbar horse-power. One might have quite a good cylinder performance but a poor boiler, and would get very poor drawbar horse-power figures in consequence.

With regard to the question of driving "on the throttle", the author had brought out very forcibly the theoretical advantage of a high boiler pressure and of using that high boiler pressure to get a big expansion ratio; this was very effectively used in the locomotives which featured largely in the paper: the converted Royal Scots. But one often saw locomotives driven, apparently with reasonable success, with a lengthy cut-off and a partly-opened regulator, and he had often wondered whether tests had ever been made to find out exactly what difference in coal consumption there was, as between running an engine with, say, a wide-open regulator and 10 per cent cut-off and running with 30 per cent cut-off and the regulator opened just wide enough to maintain the scheduled speed.

Engines were often remanned at intermediate stations, and the two drivers sometimes worked in totally different ways, and got almost equally good results. He would like to know whether tests had ever been carried out to ascertain the coal consumptions in those two cases.

Mr. H. I. Andrews, M.Sc. (Eng.), A.M.I.Mech.E., referred to the author's statement that Dalby's mean effective pressure curve was a straight line cutting the zero ordinate, and suggested that Dalby was a more careful experimenter than that; what Professor Dalby actually wrote in 1906 was that the curve appeared to be sensibly straight between the limits of 400 and 1,200 ft. per min. piston speed. It was true, of course, that Dalby calculated the mean effective pressure at zero speed, and showed that that fell on an extension of the curve, though certain intermediate results did not fit in, but so far as the probable continuation of the curve was concerned he maintained a very discreet silence.

That could very readily be understood after an examination of the two test plants which were used; both were still in existence—one at Purdue and one being preserved by the Chicago and North Western Railroad as a museum piece at Illinois. There was no doubt that very considerable skill had been needed to obtain reliable results at 50 m.p.h. on those two plants.

In the application of the suggestions contained in the present paper, if attention were diverted from the true range of those observations on which the expression in question was based, incorrect results might be obtained. The author would agree because he had elsewhere commented on a school of thought which seemed prepared to calculate the entire performance of the locomotive on the basis of a few isolated experiments. What Dalby actually did was to establish an approximate relationship over a limited range, from which he was able to deduce the maximum indicated horse-power of the engine and the speed at which it would occur. It was for that reason that the line on Fig. 9 had been extended down to the zero ordinate—as half any speed so obtained represented the speed at maximum indicated horse-power. That relationship of Dalby's had survived for some forty years; provided the relationship given in the present paper was employed in a similar manner to Dalby's, it would give correspondingly valuable results, he suggested.

* Windle, E. 1931 Jl. Inst.Loco.E., vol. 21, p. 178, "Some Notes Relating to Cylinder Performance".

As an example, however, of what might be obtained by the injudicious use of a formula of this type, he had taken the 22 per cent cut-off curve for the rebuilt Royal Scot locomotive shown in Fig. 11, and, following the procedure of Dalby, he had calculated the maximum indicated horse-power which could be obtained; and it worked out at 3887·4. There was nothing wrong with the relationship on which it was calculated, and there was nothing wrong, he hoped, with the calculation; but, in the course of the calculation, the argument had slipped out of the range of the observations upon which the relationship was based.

Mr. J. L. Wilson, G.I.Mech.E., dealing with the use of the author's basic data curves as a criterion for assessing the performance of an existing locomotive in respect of power developed, said he was new to the question of locomotive performance and was not even professionally concerned with it, but on his few footplate trips he had noted certain results to which he had tried to apply the author's method of analysis. On the L.N.E.R., between Newark and mile-post 127 the track was almost level, and with an A4 Pacific and 530 tons load a steady speed of 64 m.p.h. was reached, to maintain which, from estimates based upon resistance curves, would require a drawbar pull of 7,000 lb. The observed conditions from the footplate were 160 lb. per sq. in. steam-chest pressure at 17 per cent cut-off, for which the author's curves, characteristic and the $\sqrt{N}$ factor gave a mean effective pressure of 35·7 lb. per sq. in., equivalent to a tractive effort of only 5,900 lb. He could not find the reason for the discrepancy between a tractive effort of 5,900 lb. and a *required* drawbar pull of 7,000 lb. He found the same discrepancy for the next section, for level track with a constant speed of 60 m.p.h.

He thought that it must have something to do with the locomotive, but taking a locomotive of the Merchant Navy class, again on level track and with a speed of 56 m.p.h., with 130 lb. per sq. in. steam-chest pressure and 15 per cent cut-off, he found a similar discrepancy between the tractive effort and the required drawbar pull. He wondered whether the author could shed any light on that. If those cut-offs were increased to 25 per cent, theoretically one would get back to about the correct answer—over quite a large range of speeds which he had examined. It seemed to him that either the speed correction factor, $\sqrt{N}$, **was** wrong somewhere, or that the value P_c, derived from the basic data curves, had to be derived in some other way than by simple observation of the conditions on the engine.

He had used the Kiesel formula in a modified form to take account of the fixed cut-off at which most engines ran in these days—they were worked down to a minimum cut-off and then throttle-governed—and, by taking the steam consumption used in the formula from the engine conditions (not from the evaporation rate) obtained agreement with the power required according to calculations based upon drag drawbar pull of the locomotive. Perhaps the author's comments on the Kiesel formula should not be so severely critical if it were used in the correct manner, or rather in the way in which he himself had used it.

He did not think that the boiler pressure had anything to do with the output of a locomotive in these days; full boiler pressure was rarely used in the operating range between 40 and 60 m.p.h. The steam had to be throttled down to keep the steam consumption of the locomotive within the firing rate possible with manual firing, so that really high boiler pressures seemed to have little relation to locomotive performance.

Finally, he would like to show a simple non-dimensional method of plotting locomotive characteristics (Fig. 34) in which he plotted steam flow/pressure against i.h.p./pressure and obtained for a given cut-off (say 15 per cent) a straight line. On plotting the drawbar resistance over the 17 per cent cut-off line from Fig. 34, it would be found that (Fig. 35) the points lay together within ± 5 per cent. Similarly, one could plot efficiency, and the curve would be of the type shown in Fig. 36.

As the author had said, for a given steam flow at a given cut-off, one gained very little by increasing the steam pressure: one might lose, because the efficiency of the engine increased at lower pressure. He suggested that that was borne out by practical results (Fig. 36): it was steam-chest pressure which counted, and not the actual boiler pressure.

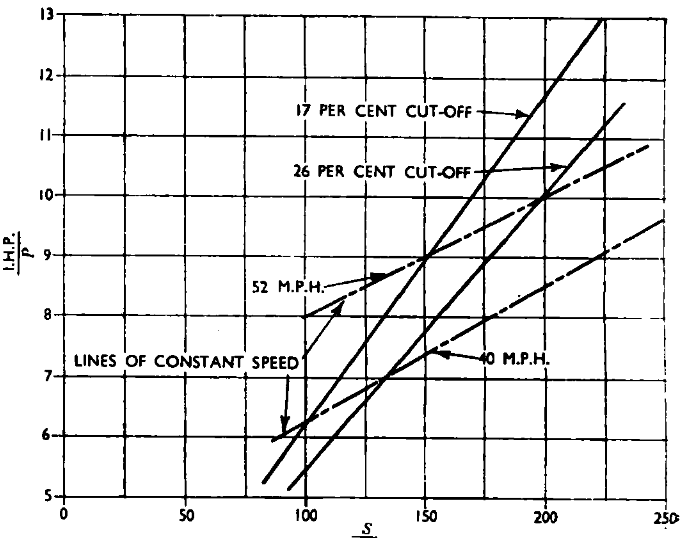


Fig. 34. Power in Operation

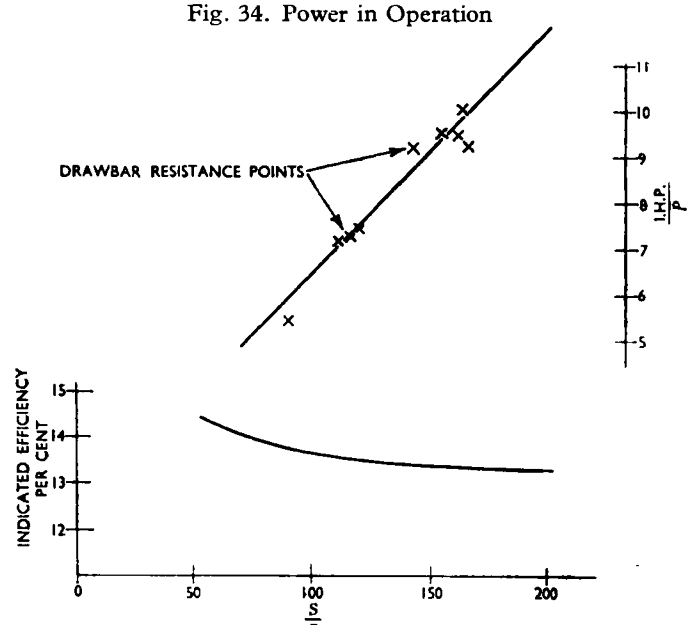


Fig. 35. Relationship of Drawbar Resistance and Indicated Horse-power/Steam Pressure

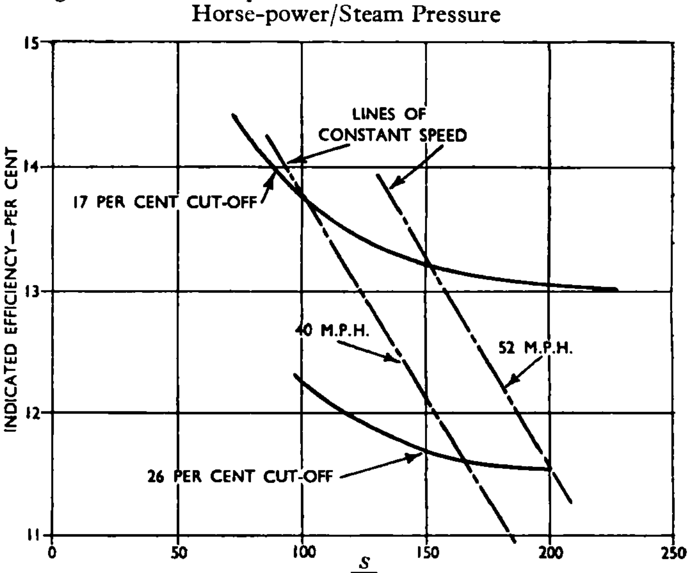


Fig. 36. Locomotive Power and Control

Mr. D. R. Carling, M.A., A.M.I.Mech.E., said that the whole of the author's thesis was really devoted to establishing a criterion for the measurement, on a consistent basis, of the extent to which any locomotive maintained its theoretical output with rising speed, and also by which one locomotive might be compared with another. The basis, however, was not a fixed one; there was a different criterion for each different boiler pressure. In other words, the author was not providing a strict measure of one locomotive against another, but rather of the relative effectiveness of one locomotive's steam circuit and distribution against those of another, making allowance for the possible difference of initial pressure of the working fluid.

Again, the actual standard of comparison was also varied according to one particular feature of the steam circuit—the clearance volume—but it was surely wrong in principle to provide a standard which varied according to the measure of one part of the article which was to be compared to that standard. The amount of the clearance volume was as much a part of the design of the steam passages as the number and size of the superheater elements or the diameter of the blast pipe, or even the characteristics of the valve gear; indeed, it was most intimately bound up with certain features of the valves and valve gear. That did not mean that the factor k was not of value—far from it—but it did mean that it was necessary to think very clearly just what k really was, before using it as a means of comparison.

The author had criticized some formulae for locomotive power which were based on boiler dimensions and assumed an engine of good current design, but was that fundamentally any less correct than basing the power on the dimensions of the engine and assuming that a boiler of good current design would provide the necessary steam? A more accurate knowledge of the rate of steam consumption might indicate more accurately the power likely to be developed from a given rate of steam production, but the upper limit would still be fixed by the boiler's ability to produce the necessary steam. Every boiler had a final upper limit beyond which no increase in rate of firing would produce any more steam, and in practice even that limit was not nearly attained. His criticism was not intended to detract from the great service that the author had rendered in making such a useful analysis of the processes which went on in the cylinders of a locomotive.

In view of modern tendencies, the data for the "calculated" indicator diagrams might well have been chosen at 700 deg. F., or even higher, and at 17 lb. per sq. in. abs., or even lower, for superheat and back-pressure respectively.

Incidentally, the curves in Figs. 3 and 4 helped to show, at least in part, why most compound locomotives with low boiler pressures had not equalled expectation. They had worked mainly on the drooping portions of the curves.

He was glad to see that Professor Nordmann's work had been assessed at its correct high value, both the paper referred to and Professor Nordmann's earlier work being of great interest. At the same time, the method of presenting the test results was rather confusing. The intention was probably to get similar power outputs from the engines at the high and low boiler pressures, and with the two- and three-cylinder engines in each case, but this had resulted in no two working at the same cut-off. Also, it was not clear whether the curves of steam consumption were obtained for conditions of maximum economy or maximum power or at a fixed rate of steam production. Once again there was justification for the use of two cylinders rather than three or four, whenever there were not really good reasons for adopting the multicylinder type of engine. Indeed, one enthusiastic Teuton had even gone so far as to propose, and design, a one-cylinder locomotive.

It was very interesting to find that the actual steam consumption figures were lower than the calculated figures, showing that the imperfections of actual locomotives, which led to loss of power, did not necessarily also lead to loss of efficiency.

Fig. 8 showed clearly how important it was to keep down back pressure to a minimum, and in that respect the best modern practice represented a great advance not only over that of earlier days but also over much of what was still in vogue.

The author had very wisely drawn attention to the fact that locomotives worked for a considerable part of their time with the regulator only partly open. It was not often realized how small a percentage of a locomotive's possible output was normally required. The author also made a good point in recalling that the actual cut-off at which a locomotive was working might vary a good deal from the nominal value, even with careful valve setting. This meant that care must be taken in accepting any quoted figures unless there were very good grounds for supposing the stated cut-offs to have been checked at all values. Unfortunately in most tests there was no indication that the valve gear had been checked. There was the further difficulty that the ordinary type of indicator became less and less reliable as speeds increased beyond 250 r.p.m. Generally the tendency was for the diagram to have a larger area than it should have.

There was a possible reason for the high power output at high speed of some locomotives fitted with poppet valves. Most poppet valves were returned to the closed position by springs or by steam pressure, and, should there be even a minute delay in closure, the nominal cut-off would be increased. If the nominal cut-off was very early and speed very high, the effect of delay in closure of the inlet valve would be very marked, while delay in closure of the exhaust valve might also make the indicator diagrams fuller, provided that the inlet valve opened early enough for boiler steam to fill the clearance spaces. As most poppet-valve gears gave a quick opening, this might usually be taken to be the case. At exceptionally high speeds, poppet valves might even bounce off their seats after closing.

He had not been able to compare many direct test results with values derived from the author's formulae, due to a lack of indicator diagrams (for any reasonably modern locomotives)

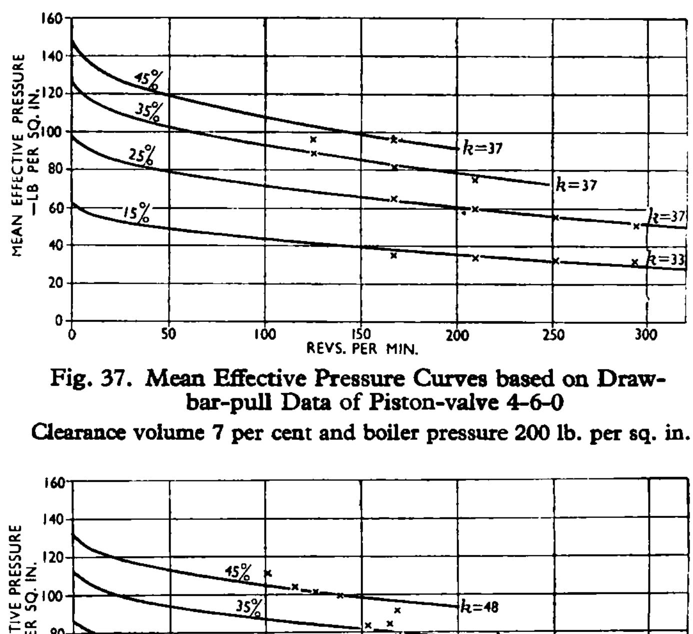


Fig. 37. Mean Effective Pressure Curves based on Drawbar-pull Data of Piston-valve 4-6-0

Clearance volume 7 per cent and boiler pressure 200 lb. per sq. in.

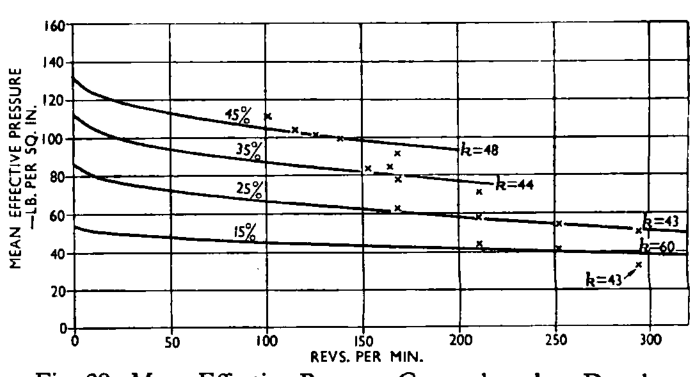


Fig. 38. Mean Effective Pressure Curves based on Drawbar-pull Data of Piston-valve 4-4-0*

Clearance volume 7 per cent and boiler pressure 180 lb. per sq. in.

taken at high speeds, but he had tried the effect of deducing mean effective pressures from some very well-established drawbar-pull values, obtained in circumstances known to him, by using a suitable formula for the resistance of engine and tender.

The points thus obtained agreed with curves of the suggested type as well as most of the points shown in the paper.

* L.N.E.R. three-cylinder 4-4-0 class "D.49" (Shire).

A 4-6-0* designed about 1928 gave a very consistent factor k of 37, between 45 per cent and 25 per cent, but of only 33 at 15 per cent cut-off (Fig. 37).

Conversely, a 4-4-0 piston-valve engine, designed about 1927, gave a k dropping from 48 to 43 as the cut-off was shortened from 45 per cent to 25 per cent, but as high as 60 for 15 per cent, up to 250 r.p.m.: if, however, preference were given to an r.p.m. of 300 the value of k for 15 per cent dropped to 43 (Fig. 38).

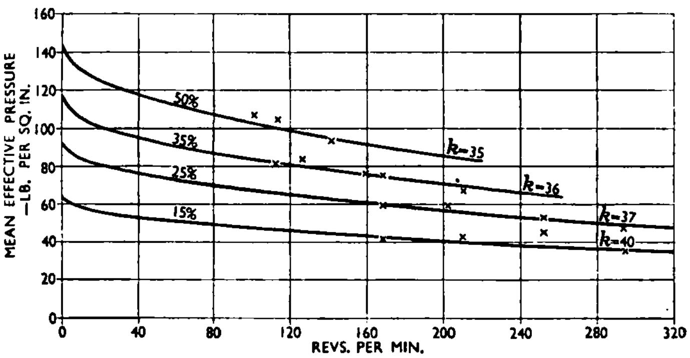


Fig. 39. Mean Effective Pressure Curves based on Drawbar-pull Data of 4-4-0 fitted with Rotary Lentz Valve Gear

Clearance volume 11½ per cent and boiler pressure 180 lb. per sq. in.

An identical 4-4-0 fitted with poppet valves gave a k rising from 35 to 40, as the cut-off came down from 50 per cent to 15 per cent. This engine also showed a marked drop above 250 r.p.m. in 15 per cent, and to a less extent in 25 per cent cut-off (Fig. 39). The values of k for the poppet-valve engine were all lower than those for the piston-valve engine—just the reverse of the examples quoted in the paper; this might be attributable to the fact that the poppet valves required a clearance volume 60 per cent larger than the piston valve.

* L.N.E.R. three-cylinder 4-6-0 class "B.17" (Sandringham).

Mr. E. L. Diamond replied that, pending fuller consideration of the discussion, he would deal briefly with three points only. First, he deprecated the idea that he had proposed in any way a new standard cycle. His theoretical diagrams were merely calculated in accordance with classic thermodynamic theory; and so far from having used them as a standard of comparison, he had referred his curves of relative efficiency to the Rankine cycle, which he regarded as the most appropriate standard.

Secondly, he wished to make it clear that his method of calculating horse-power was not intended to replace methods aimed at determining the ultimate power capacity of the locomotive as limited by the boiler. For the economical working of locomotives at speed it was necessary to use a better expansion ratio than would normally be required to exhaust the boiler. An express locomotive for which this was not true was not well designed. As Mr. Cox had said, time tables should not require the locomotives to run at the point of just draining the boiler. Further, it was quite fallacious to suppose that the ultimate boiler capacity was in practice a constant. It varied widely with the quality of the fuel, the skill of the fireman, and above all the condition of the locomotive, and it would be most unreliable as a basis for calculating train schedules. It was of practical value to have some means of determining accurately the power available at a specified and suitable cut-off well within the boiler capacity.

Dr. Johansen and Mr. Holcroft had suggested that mean effective pressure diagrams should perhaps be plotted on a basis of piston speed rather than of revolutions per minute. The stroke in most modern express locomotives was very much the same, and Mr. Holcroft's railcar example lay somewhat outside the scope of the paper, but the point was a good one, and he would consider it further.

Communications

Mr. C. A. Cardew (Sydney, New South Wales) wrote that the matters dealt with had great influence, both on the operating capacity (throughout the range of running speeds) and on the thermal efficiency of the steam locomotive. The author's central theme was, in effect, the diagram factor of the steam-engine indicator card, as applied to locomotives, and more especially the sort of factor obtained, or obtainable, from various designs and types of locomotive engines when running at fairly high speeds.

The author suggested that locomotive engineers of recent years had laid too little emphasis on the capacity of the cylinders of the same locomotives, for utilizing, to the fullest extent (and with the utmost economy), the power potential of the boiler. He was in some agreement with the author in this contention; the locomotive passed, in much earlier days, from a state in which cylinder capacity was primarily the consideration, and boiler capacity, in relation thereto, was largely neglected, to a later era of the dominance of the boiler, dating perhaps, in British practice, from the introduction of the several "Dunalastair" classes on the Caledonian Railway between 1896 and 1904. This paper should serve, perhaps, to readjust the balance between the attention given to the boiler and that to the cylinders—particularly to the ability and efficiency with which these latter could both be supplied with, and discharge, ample quantities of steam.

He himself considered that every steam locomotive had, in fact, two characteristic, or tractive effort/speed curves applicable to it. One corresponded to the maximum horse-power which such a locomotive could maintain for long periods, and this was usually dependent on a limit determined by the capacity of the boiler for sustained evaporative capacity, with the fuel employed. The other (which could often be maintained, in practice, for a limited period only—because of boiler limitations) was chiefly fixed by the capacity of the cylinders for maintaining the highest possible mean effective pressure, at high rates of steam flow. Other factors which might affect output, however, were limits fixed by adhesion, and the possibility of excessive smokebox draught reacting too severely on the fire.

The first characteristic curve was dependent on continuous boiler rating and, in a well-designed locomotive, should be attainable, as the author later contended, with a cut-off in the cylinders which was somewhere in the region of the optimum for economical cylinder performance, and with such firing rates, and other boiler conditions, as made for the maximum economy in steam generation as well. The second, being required for short distances and periods only, need not, and could not well, take so much account of economical working factors; it was a legitimate expedient for dealing with circumstances encountered in the operation of the railway concerned only at some few points.

The author spoke of the practicability of operating some of the older designs in the full-gear position, and with wide-open regulator, at speed. Such locomotives would, presumably, have been designed, say, forty to fifty years ago, when the era of the boiler as the paramount consideration in locomotive design was beginning, and the over-cylindering, which was common some decades earlier, was going out of vogue. Whilst his own locomotive operating experience could not support the author to the fullest extent, he was able to endorse the fact that very late cut-off working, up to surprising speeds, was within the capacity for continuous steam supply of the boilers of many locomotives of the period mentioned. But, was it not possible that, with examples of which the author had had experience, the diagram factor applying at speed was so poor, more particularly on account of restricted port opening to steam admission, that, as a result of the wire-drawing, there was produced, in effect, a point of cut-off in the cylinders much earlier than that nominally provided by the valve events. Also, as the speed increased, the amount of steam which could enter the cylinders in the period of time available would, on account of wire-drawing, be curtailed, and because of this, as well as of the measure of expansive working actually taking place, the steam demand might not have been so excessive as might otherwise be made to appear. That state of affairs could exist with some older locomotive designs, but hardly with well-designed modern locomotives, and it would, therefore, be impossible to operate the latter, at any considerable speed, in the manner which the author had described, though this would be no reflection on the boiler of the modern engine.

He was interested in the author's comments on the effect of clearance volume on the cylinder performance of the locomotive; the clearance volume, naturally, played a part of increasing importance as cut-off was shortened. The graphs, however, were based on certain assumptions and they took no account of such effects as varying leakage quantities, radiation, condensation, and wire-drawing during admission; the author intended, no doubt, that those should be kept in mind.

In connexion with the effect of clearance volume causing loss of relative cylinder efficiency at high pressures, he suggested that, if the greater weight of steam lost owing to leakage at higher pressure, were to be taken into account (e.g. at the piston valve rings all the time the engine was working, but at the piston rings mostly during the period of admission), there might actually be an overall net loss of efficiency chargeable against the higher steam pressure, as compared with pressures which were more moderate. Especially might this be so after the locomotive had run a high mileage. Thus leakage and its effect on the relative steam consumption of the two simple expansion engines which were tested under high and low steam-pressure working conditions by Professor Nordmann, might have been one of the factors accounting for the failure of the engines, when working under the higher boiler pressure, to achieve any marked superiority in efficiency compared with the lower working pressure conditions.

Regarding the tendency to the use of higher steam pressures in locomotive practice, the author appeared to attribute this to the desire to obtain the required power output with cylinders limited by the loading gauge, or to avoid the introduction of multi-cylinder locomotives. He himself thought there was more justification for higher pressures on those grounds than on the score of steam economy.

In Fig. 12, the comparison afforded by the indicator cards for the two L.M.S. locomotives, typical of the locomotive practice of the years 1917 and 1944 respectively, were of interest, particularly as regards the nearer approach to the hypothetical of the exhaust and compression lines in the diagram applying to the rebuilt Royal Scot locomotive of 1944 design. On the admission side, however, it was disappointing to note that the peak admission pressure was some 35 lb. per sq. in. below boiler pressure, with the Royal Scot, although only some 15 lb. per sq. in. lower in the case of the earlier engine. Cards taken with the same boiler pressure and rate of steam flow would be more strictly comparable.

The values shown in Fig. 13 did not always truly reflect the relative merits of different types of valves, as media for providing the admission to and rejection from, the cylinders of a locomotive. The nominal areas of the steam and exhaust port openings, particularly in the case of some piston-valve arrangement, could often fail to provide the advantages which they appeared to possess, by reason of conditions other than those existing at the port faces. Thus, other parts of the steam and exhaust passages might be less favourable to steam flow on account of local restrictions, or parts of some passages (such as those surrounding the ordinary piston valve) might have areas (in this case those in the region remote from the cylinder) which could not be used to the best advantage.

The comparative results as to the mean effective pressures actually realized with various piston and poppet type valves, as determined by tests on the French Railways and on the Pennsylvania Railroad (Figs. 14 to 19 inclusive), were striking. The author claimed that, except in the case of Fig. 18 for the Pennsylvania Railroad K4S class locomotive, each curve lay very close to a path running through the mean of the values represented by the points as plotted from the test results. This, in general, was so, but, in all the curves exhibited in Figs. 15 to 18 (inclusive), there were no test-result points given for speeds of less than 120 r.p.m.; in Fig. 19 none for speeds lower than 150 r.p.m., and, in Fig. 14, lower than 100 r.p.m. Therefore, although the curves fitted the test data well in the higher speed range, there was no check on results at lower rates from test data, and it was just possible that, in this lower range, exact agreement did not hold. In the case of Fig. 14, the author himself commented on a tendency for the test results to depart from the curves with late cut-off, at the lower speeds, ascribing for the phenomenon certain reasons. He himself was not quite clear as to why this departure was apparent at the lower speeds and longer cut-offs, only, unless the constant k, in the characteristic power equation, chosen as a suitable function to employ for computing results from the particular valve and valve gear involved, at early cut-off, with high speed, was not equally applicable when the engine was worked with a late cut-off and at low speed.

He had conducted tests in which the same locomotive was fitted first with ordinary piston valves, and subsequently with piston valves of the double-admission type, no other change being made other than the substitution of the double-ported valve, with its valve liners, for the original piston valve, etc.

With this 4–6–0 express locomotive, having driving wheels 5 ft. 9 ins., boiler pressure 180 lb. per sq. in., and cylinders 23 inches × 26 inches, as tested on heavy passenger train services, operating over long stretches of 1 in 75 grades, and in certain instances negotiating inclines as steep as 1 in 33 and 1 in 40, over routes ranging from some 80 to 300 miles in length, there was a saving (the average form all the tests) for the double-admission type valve as compared with the standard valve, of 8·34 per cent in coal and 6·7 per cent in water. Further, on an uphill section, some 8·35 miles long, 5·2 miles being at 1 in 75, and the remainder at 1 in 88 to 1 in 132, rising in each case, with a load of 288 tons behind the tender, the following were the running performances with each type of valve on the section Bargo to Yerrinbool, of the Southern Line, of the New South Wales Railways.

Type of piston valve	Regulator position	Mean cut-off, per cent	Time occupied	Average running speed, m.p.h.
Standard .	¾ open	42	17 min. 45 sec.	28·2
Double-admission . .	¾ open	37·8	15 min. 42 sec.	31·9

The above performance needed little comment except, perhaps, to call attention to the fact that it was a good illustration of the way in which, by the attainment of a better diagram factor, a greater cylinder power output might be realized with earlier cut-off.

The two indicator superimposed diagrams (Fig. 40) were both from the crank end of the cylinder, and represented the two types of valve, but taken with speed, cut-off, regulator opening, and boiler pressure, as nearly as possible identical in each case. The gain in card area and the better definition in the point of cut-off, with the card for the double admission type valve, was very apparent.

With reference to the proposed formula converting P_c into P_n, the formula was really intended as a means of deriving the diagram factors appropriate to certain combinations of speed and cut-off, and, in this respect, provided it could be relied upon for doing so under all the widely varying conditions which would affect the matter with different locomotive designs, and types, and conditions of operation, its value was undoubted. An element of doubt arising in regard to this concerned the values of k, and the ability of designers to establish such values accurately beforehand, for untried designs. The gaps between the values of the suggested constants for various valve and valve-gear conditions were considerable. No doubt many designs of locomotives had constant values falling within the gaps, and differing from those shown; the degree of accuracy attainable from the values of k listed was probably not great.

The author had throughout used, as a speed basis, the rotational velocity of the driving wheels, rather than the mean piston speed employed by Cole and others; there might be loss in accuracy, and in the value for comparative purposes, of the formulae and curves presented, if they were applied to widely differing locomotive designs. At a specific cut-off, the factor generally influencing the mean effective pressure in the cylinders of a given engine would seem to be the ratio between the linear velocity of the valve, at any given time, and the rate at which volumetric displacement was caused by the movement of the piston in the cylinder—a ratio which was not truly taken into account by the rotational velocity. The objection applied especially if application were made to several locomotives having considerable differences in piston stroke. True mean piston speed was not a completely satisfactory speed basis; even the rate of volumetric displacement due to piston movement, though nearer the ideal, did not entirely fulfil requirements; but the employment of one of these in place of revolutions per minute seemed preferable. He suggested, therefore, that, in the formula proposed for the estimation of the mean effective pressure at given cut-off, the speed factor should be in terms of, say, cu. ft. of piston displacement, per minute, instead of in revolutions per minute. This would leave an appropriate constant k, to cover only the valve, valve gear, and steam-port peculiarities of different locomotive designs, and would remove the necessity for any chosen values of k having to embrace, additionally, differences in lengths of piston stroke, or other cylinder volume which might, to some appreciable degree, affect the results for locomotives having widely differing cylinder proportions. The author had proceeded somewhat along these lines in the case of Fig. 12, in that he had drawn the two indicator diagrams there submitted on the basis of swept volume, whereas the suggestion made regarding the basis for the other data just mentioned was, of course, volume swept by the piston in cu. ft. per min.

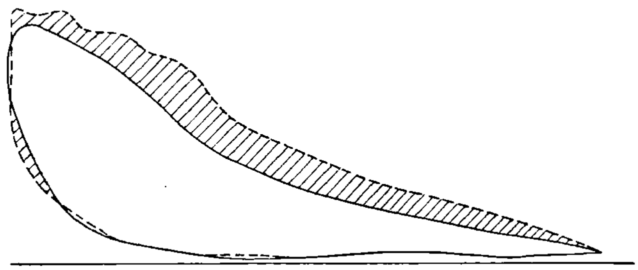


Fig. 40. Comparison between Indicator Diagrams obtained with Standard and Double-admission Type Piston Valves

Cut-off 30 per cent, regulator half open, boiler pressure 180 lb. per sq. in., speed 30 m.p.h., for both valves. Admission area 0·77 sq. in., expansion area 0·45 sq. in., total area 1·22 sq. in. for standard valve. Admission area 0·96 sq. in., expansion area 0·70 sq. in., total area 1·66 sq. in. for double-admission valve. (Increase 24·7, 55·5, and 36·1 per cent respectively.)
Indicator spring scale = 120 lb. per sq. in. per inch.

The author suggested that a good designer would arrange for the maximum continuous horse-power output required of his engine to be exerted at the optimum point of cut-off, for the steam pressure used, and so secure the most economical cylinder performance when working under these conditions. Provided that the maximum continuous horse-power output was required for a large portion of the total operating period of the locomotive in road service, this was sound in principle, though difficult to attain in practice—but this was not always the case. For example, on the railway with which he was associated the maximum continuous horse-power output of the express locomotives was generally required on long rising grades—perhaps unbroken stretches of some 30 miles in length, mostly inclined at 1 in 75. If the locomotives concerned were designed so that the cut-off, with the existing steam pressure, was shortened from 35 or 40 per cent to 25 or 30 per cent, material economy would result. But this would require a larger cylinder, which would correspondingly increase the tractive effort realized at starting, and very low speeds, thus diminishing the adhesion factor to such a low value in most cases as to make it very difficult to handle the engines under these conditions without the occurrence of violent slipping. As the engines concerned, on the same journey, often had to operate the same train loads over several shorter lengths of 1 in 40 grade, with speed approximately 12 to 15 m.p.h., this factor would have to be reckoned with. The problem might be dealt with, perhaps, on the limited cut-off principle, so as to curtail, in some measure, the available tractive effort at low speeds, though this was not an entirely satisfactory solution for starting conditions: less even torque resulted, at least with two-cylinder engines, and less effective use was made of the available adhesion.

On such a locomotive some means of pressure control was required, either separate from, or working in conjunction with, the present regulator, and by which the pressure of the steam supplied at the steam chests could be reduced, at will, to a certain fixed lower value—so that starting, and very low speed operation, could be accomplished with low-pressure conditions applying, keeping the tractive effort within the range suitable for the adhesion provided, though permitting fairly late cut-off to preserve a better turning moment on the cranks than early cut-off afforded. Such a device would need to be of the nature of a pressure-reducing valve, allowing pressure to increase gradually as speed rose and cut-off was shortened, until, at a certain cut-off, full boiler pressure was brought into operation.

In effect, the curves in Fig. 20 were an indication of the diagram factors obtainable at the various speeds, and they purported to show, on a truly comparative basis, the relative capacity of the locomotives concerned for approaching the "basic" mean effective pressure value. This they did, but they were no criterion of capacity for sustained maximum performance, which must depend on the relation of cylinder demand to boiler capacity.

The author emphasized that the constant k concerned the power/weight ratio of the locomotive at high speed, without being any direct index as to its thermal efficiency. He agreed with the contention that a high power/weight ratio was generally of more importance, in railway operation, than a high thermal efficiency but, as the two were related, locomotive engineers should seek a high k for all designs, as a means towards securing both. The writer would not even except shunting and banking engines, for, though it might not be essential to have a high value of this constant for such engines, the ability to preserve a good diagram factor, with early cut-off, would enable these engines to be worked with a greater expansion ratio than usual, with economy as a result. The high capacity for acceleration, which a good value for k should confer on engines such as those last mentioned, seemed to be a feature particularly to be desired for the class of service involved

Monsieur A. Chapelon (Paris) wrote that the curves of efficiency and of steam consumption per i.h.p.-hr. enabled one to gain an idea of the effect, good or bad, of the various factors at play, and notably of the increase of steam pressure which demanded, in order to obtain a clear advantage, the use of an early cut-off and, above all, small clearance spaces.

The author took as a condition for the establishment of the diagram at zero speed an exhaust pressure of 18 lb. per sq. in., but he did not take account of the loss due to the exhaust advance—which was considerable at low speeds. Table 4 recorded the results obtained.

In considering the pressure of 17 kg. per sq. cm. for example, it was thus seen that the effect of advancing the exhaust at zero speed *reduced* the mean effective pressure in the proportion:—

1 to 0·851	for a cut-off of	50	per cent
1 to 0·811	,, ,,	30	,,
1 to 0·725	,, ,,	10	,,

These differences appreciably affected the values of P_c and k. As the emptying of the cylinder was not instantaneous, the actual effort at low speeds was perhaps a little greater than that thus derived.

The coefficient k was determined by taking for P_c a *theoretical value*, and for P_N *a value actually measured*. Many errors could arise in the determination of P_N, whether in consequence of the deformation of the diagrams under the influence of speed, of the length of piping connecting the indicator to the cylinders, of the degree to which the system was exhausted after taking each diagram, of error in calibrating the spring of the indicator, or in the value accepted for the diameter of the piston. Was it permissible then to draw a curve through a theoretical point

TABLE 4. RATIOS OF MEAN EFFECTIVE PRESSURE TO ADMISSION PRESSURE

Cut-off, per cent	Without exhaust advance				With exhaust advance*		
	As given by the paper for 300 deg. C. (572 deg. F.), 17 kg. per sq. cm. (242 lb. per sq. in.); back pressure, 18 lb. per sq. in.	As given by the Mollier diagram for 320 deg. C., (608 deg. F.) back pressure nil, admission pressure:—			As given by the Mollier diagram for 320 deg. C. (608 deg. F.) back pressure nil, admission pressure:—		
		12 kg. per sq. cm. (171 lb. per sq. in.)	17 kg. per sq. cm. (242 lb. per sq. in.)	20 kg. per sq. cm. (285 lb. per sq. in.)	12 kg. per sq. cm. (171 lb. per sq. in.)	17 kg. per sq. cm. (242 lb. per sq. in.)	20 kg. per sq. cm. (285 lb. per sq. in.)
50	0·805	0·802	0·802	0·802	0·686	0·686	0·686
30	0·593	0·592	0·615	0·595	0·487	0·497	0·483
10	0·267	0·257	0·297	0·2925	0·184	0·2105	0·206

Clearance space, 10 per cent.

* The case of a normal Walschaert's gear.

and an experimental point, or would it be better to compare only experimental points to study how the tractive effort varied with the speed?

The author's conclusions in regard to the effect of the increase of steam pressure were pessimistic. Such an increase of steam pressure was emphatically not accompanied by exclusively favourable factors, even in the case of the theoretical engine, i.e. in the absence of all phenomena connected with the action of surfaces and of leakage which profoundly affected the functioning of an actual engine, as Hirn and his pupil had demonstrated a long time ago. The author's study had the merit of showing up clearly the great influence of the clearance space on the efficiency, at least in simple expansion engines—an influence which became greater and greater as the steam pressure

TABLE 5. ECONOMY WITH HIGH PRESSURE

Speed, km. per hr.	No. 84002 (3 cylinders)	No. 84003 (2 cylinders)
Economy by using 20 kg. per sq. cm. as against 14 kg. per sq. cm., per i.h.p.-hr., per cent		
30	12·7	14·8
40	10·2	12·4
60	4·3	6·4
Economy at 20 kg. per sq. cm. per d.h.p.-hr., per cent		
30	11·10	10·7
40	6·41	8·25
60	0·89	0

increased. Increase of pressure was nevertheless a means of increasing the efficiency of engines, when this effect was taken correctly into account.

The terms of comparison had to be fixed definitely, or one might draw conclusions entirely incorrect.

The trials of Professor Nordmann on the two locomotives 84002 and 84003 of the Reichsbahn, of three and of two cylinders respectively, and of type 2–10–2 showed (Table 5) the economy in favour of the pressure of 20 kg. per sq. cm. (285 lb. per sq. in.) as against that of 14 kg. per sq. cm. (200 lb. per sq. in.).

The economy in respect of the drawbar work was a little less than that in respect of the work in the cylinders, but this economy was none the less apparent, above all at low speeds.

It was clear from an examination of the ratio P_u/P_i of the drawbar h.p. and the i.h.p. that, in passing from the pressure of 14 kg. per sq. cm. to that of 20 kg. per sq. cm., this ratio was reduced:—

(1) At 30 km. per hr. (18·7 m.p.h.) in the proportion of 1 to 0·98 for the three-cylinder locomotive, and of 1 to 0·95 for the locomotive with two cylinders.

(2) At 40 km. per hr. (24·9 m.p.h.) in the ratio of 1 to 0·97 for the three cylinders, and of 1 to 0·95 for the three cylinders.

(3) At 60 km. per hr. (37·4 m.p.h.) in the ratio of 1 to 0·97 for the three cylinders, and of 1 to 0·933 for the two cylinders.

The reduction in this ratio increased very little with the speed as the pressure increased; this disproved the hypothesis that it was the increase of the mechanical resistance with the speed which annulled the economy due to the pressure of 20 kg. per sq. cm. at 60 km. per hr. In reality this was due to other causes, and probably to a double error affecting at the same time the indicator diagrams and the recording of the tractive effort by the dynamometer car.

The curves of indicated horse-power as a function of speed, published by Professor Nordmann, showed a less rapid increase with the speed for 20 kg. per sq. cm. than for 14 kg. per sq. cm. This might be because the more attenuated diagrams at the early cut-offs, with a pressure of 20 kg. per sq. cm., had been more deformed and more inflated at low speeds than those at the longer cut-offs, with a pressure of 14 kg. per sq. cm., in consequence of the influence of the greater humidity of the steam and perhaps of insufficient exhaust of the indicator piping.

The drawbar horse-powers could be diminished at high speed in consequence of the oscillations due to the driving couple, which would be greater at the short cut-offs used at a pressure of 20 kg. per sq. cm. than at the longer cut-offs used at 14 kg. per sq. cm.

Hence the effect of the speed made itself felt more on the two-cylinder locomotive, with its more irregular couple, than on the three-cylinder—perhaps for the same reason.

Such an explanation had been envisaged in consequence of many trials on the testing plant at Vitry; these demonstrated the great importance of the longitudinal oscillations of the locomotives on the value of the drawbar pulls recorded when special steps had not been taken, by other means, to equalize the effort inscribed on the dynamometer record.

It was evident that a locomotive running at a cut-off of 30 per cent would set up oscillations appreciably stronger than a locomotive running at a cut-off of 45 per cent, as was the case for the two German locomotives considered. In fact one could accept the results of the German trials only with the greatest reserve. The beneficial effect of the high pressure was, on the contrary, appreciably greater at high speeds, where the loss of pressure became a dominant factor, and where the neutralizing effects of surface losses and leakage at the high pressures tended to disappear.

His own department had, for a long time past, employed numerous measures to reveal the value of high pressures, or—which amounted to the same thing—the value of running at full regulator opening; the results had always been favourable. The results in Table 6 had been obtained in testing Pacific locomotives, with saturated steam and with superheated steam, running with varying pressures in their steam chests.

The economy due to the full open regulator was thus from 20 per cent in water and from 14·7 per cent in coal, for the superheated machine, and from 21·4 per cent in water and 19 per cent in coal, for the saturated machine.

The same trial was repeated on a simple-expansion machine, with a degree of superheat equal to that of the Pacific type, and gave an economy of about 5 per cent per drawbar horse-power-hour for an increase of steam-chest pressure from 8·5 to 10 kg. (142 lb. per sq. in.) approximately—and that in spite of intolerable vibrations which occurred with these machines at speed as soon as the steam-chest pressure passed 7–8 kg. per sq. cm. (100–114 lb. per sq. in.).

The most recent trials carried out at Vitry on a P.L.M. Mikado compound locomotive with modified steam circuit and high superheat gave, at 70 km. per hr. (43·6 m.p.h.), the results in Table 7:—

TABLE 6. ECONOMY WITH FULL REGULATOR OPENING

Regulator position	Least steam chest pressure, kg. per sq. cm. (lb. per sq. in.)	Least cut-off high-pressure and low-pressure, per cent	Least i.h.p.	Water consumption per i.h.p.-hr., kg. (lb.)	Coal consumption per i.h.p.-hr., kg. (kindling deducted) (lb.)
Superheated Locomotive.					
1. Throttled regulator	8·6 (123)	60/70	1,120	9·5 (21)	1·247 (2·76)
2. Regulator fully open	13·5 (192)	44/62	1,350	7·6 (16·8)	1·064 (2·35)
Saturated Locomotive.					
1. Throttled regulator	9·09 (130)	60·5/70	858	14·0 (30·9)	1·69 (3·73)
2. Regulator fully open	14·85 (212)	39·5/65	1,160	11·0 (24·3)	1·37 (3·03)

TABLE 7. ECONOMY WITH HIGH PRESSURE

Drawbar horse-power	Water consumption per drawbar h.p.-hr. at a pressure of 10 kg. per sq. cm., kg. (lb.)	Water consumption per drawbar h.p.-hr. at a pressure of 15 kg. per sq. cm., kg. (lb.)	Economy at pressure of 15 kg. per sq. cm. as compared with 10 kg. per sq. cm., per cent
1,500	7·6 (16·8)	6·7 (14·8)	11·8
2,000	7·3 (16·1)	6·5 (14·4)	11·0

Other tests carried out very recently with American-built 2–8–2 locomotives with two simple-expansion cylinders, also at Vitry and with steam pressures of 10 and 15·3 kg. per sq. cm. (142 and 214 lb. per sq. in.) gave the results in Table 8:—

TABLE 8. ECONOMY WITH HIGH PRESSURE

Drawbar horse-power	Water consumption per drawbar h.p.-hr. at a pressure of 10 kg. per sq. cm., kg. (lb.)	Water consumption per drawbar h.p.-hr. at a pressure of 15·3 kg. per sq. cm., kg. (lb.)	Economy at pressure of 15 kg. per sq. cm. as compared with 10 kg. per sq. cm., per cent
At speed of 40 km. per hour (24·9 m.p.h.):—			
1,050	7·5 (16·6)	7·5 (16·6)	0
1,400	8·0 (17·6)	7·25 (16·0)	9·35
At speed of 80 km. per hour (49·8 m.p.h.):—			
1,350	7·5 (16·6)	7·5 (16·6)	0
1,700	7·9 (17·4)	7 (15·4)	11·35
2,000	8·7 (19·2)	6·8 (15·0)	21·80

At powers less than those for which the economy due to the high pressure was nil, the pressure of 10 kg. per sq. cm. gave a slight advantage of the order of 5 per cent, for example, at 800 h.p., as compared with 15·3 kg. per sq. cm. This result was probably due to the influence of the effects of surface losses and, above all, leaks which handicapped the high pressures when the power developed was low and also because the great expansion ratios involved increased thermal losses in the cylinders.

The Value of k. In order to find why the coefficient k of locomotive 4701 with a superior steam circuit to 3705 were found *inferior* to those of the latter, the validity of the method, by which the mean effective pressure at zero speed was calculated, was investigated. It was found that it was not valid for a compound locomotive:—

(1) Because the clearance space of the equivalent simple-expansion locomotive was not equal to the high-pressure clearance space of the compound divided by the ratio of the volumes of the low-pressure cylinders to the high-pressure cylinders, but much less.

(2) Because the low-pressure clearance space, when the compression was insufficient, introduced an additional loss in the mean effective pressure of the theoretical locomotive (Fig. 41).

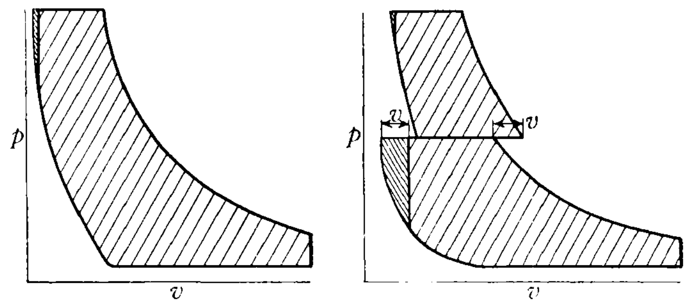


Fig. 41. Theoretical Indicator Diagrams for Simple and Compound Expansions

Compression does not completely fill the clearance volume of the high-pressure cylinders up to steam-chest pressure, and loss results as represented by the small shaded areas. A similar effect in the low-pressure cylinder causes the expansion line to be set back to the left by an amount v; additional loss is represented by the relevant shaded area.

For example, in the case of No. 4701, a cut-off of 53 per cent in the high-pressure cylinder would be equivalent to one of 25 per cent in the corresponding simple-expansion locomotive; the equivalent clearance space of the compound machine, *using the same weight of steam per cylinder* would be 4 per cent and not 11·8 per cent (25/2·10 = 11·8 per cent). In consequence of this, the mean effective pressure of this machine, in the circumstances considered, would be 137·8 lb. per sq. in. instead of about 154 lb. per sq. in.

In fact the compound only gave 131·8 lb. *as a maximum* for a low pressure cut-off of 31·5 per cent—which avoided truncation of the expansion in the high-pressure cylinders and corresponded to maximum efficiency.

The mean effective pressure of the compound would therefore be reduced to 0·955 of that of the equivalent theoretical simple expansion having a greatly reduced clearance space calculated as above.

When the coefficient k was recalculated using the figures obtained for P_N at the mean speed of 240 r.p.m., the values of k varied between 87·5 and 220 for No. 4701 at cut-offs from 28·4 to 9·5 per cent, and between 85 and 294 for No. 3705 at cut-offs from 25 to 8·6 per cent.

The difference in the high-pressure clearance volume existing between the two compound locomotives could not be an acceptable explanation of the differences found, since the clearance space of an equivalent simple-expansion locomotive was already extremely small at 4 per cent—for a high-pressure clearance space of 25 per cent. This emphasized the advantage of compounding, which permitted the adoption of the largest possible cross-sectional passage areas across the steam ports, without entailing excessive clearance space.

It was difficult to believe that the law of decrease of the mean effective pressure, was exactly an exponential one. The decrease of effort with the speed actually arose exclusively from the losses of pressure in the steam circuit, on the admission side as well as on the exhaust side. Thus the losses of pressure depended on the speed of the steam through the ports. It had been the custom to assume that these losses increased as the square of the speed:

the mean effective pressure would be expected to decrease more quickly than the speed increased, thus giving a parabolic rather than an exponential or hyperbolic curve.

From an examination of trial results of numerous locomotives, it seemed that the straight line (1) in Fig. 42, envisaged by Dalby, was one of the most frequent, notably for superheated locomotives, the parabola (2) with curvature turned towards the axis of speed being more general in the case of saturated locomotives. The exponential curve (3) which had actually been obtained in certain cases was due, he believed, either to the consequences of errors in measurement, or to an increase in the cut-off, or an increase in the degree of superheat, with speed.

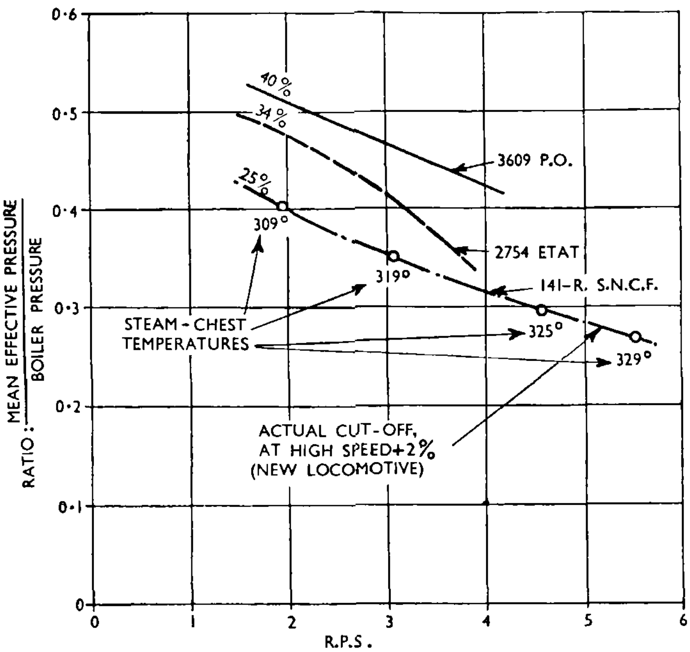


Fig. 42. Results of Tests upon the Variations of Mean Effective Pressure

In fact, his department's tests had shown that, on machines with piston valves, the throw of the valves increased in amplitude with increase of speed owing to play, and that this gave rise to an appreciable increase in cut-off with the speed. The position of the reversing lever in the cab was not a true indication of cut-off.

Superheat caused further improvement at high speed because, for a given cut-off, the quantity of steam increased with the speed, and the degree of superheat increased with the rate of flow of steam. The points on the curve at very high speed corresponded thus to diagrams with considerably increased superheat and reduced losses of pressure.

Finally, the speed was liable to falsify the indicator diagrams, especially since the piping connecting the indicators to the cylinders was not particularly short and the transmission of the movement of the piston to the indicator drum was itself subject to backlash.

Actual performance of the locomotives was a complex problem and necessitated rigorous measurement of all the factors which could influence the phenomenon.

The author had taken the value $n = 1{\cdot}33$ for the law of expansion and $n = 1{\cdot}2$ for the law of compression.

If one calculated the value of n, for various conditions of temperature and pressure up to 20 kg. per sq. cm. (285 lb. per sq. in.) and 400 deg. C. (752 deg. F.) from the Mollier diagram, one found the same laws for expansion and compression: the exponential n was approximately $1{\cdot}3$.

In the case of the compression, experiments carried out at the University of Liège, notably by Duchesne, as well as those made by other experimenters, had shown that, even in engines working with saturated steam, the steam remaining in the cylinder at the moment when compression commenced was always a little superheated in consequence of the heat retained by the walls—received from the steam during the phases of pre-exhaust and exhaust. The compression of this superheated steam followed the same law as its expansion, with the same value of the index n in the formula pv^n = constant.

Mr. C. W. Clarke, M.I.Mech.E., wrote drawing the author's attention to two papers on this subject: those of Fry* and Da Costa†. Fry had proposed a cycle which he called the "locomotive cycle", and Da Costa the "ideal cycle". Da Costa's cycle was, in fact, the cycle used by the author.

The basis of all such papers was that cylinder efficiency

$$\xi = \frac{\text{Work done per stroke}}{\text{Weight of steam used per stroke}}$$

In a reciprocating steam engine, for a given admission pressure and cut-off, compression was the only variable, and that compression which gave the maximum value of the ratio area of indicator diagram/indicated weight of cylinder feed, gave the highest cylinder efficiency. However, in practice it was necessary to take into consideration the "missing quantity" which, even in a good modern design, could exceed 15 per cent of the cylinder feed. Since the missing quantity always formed part of the denominator of the equation, the effect of compression was masked when such factors were considered, especially as the missing quantity percentage increased as the ratio of expansion increased.

He was not in agreement with some of the remarks made by the author on poppet valves and piston valves, and he felt that experimental data was needed to substantiate certain remarks. Why should a double-beat form of valve be inherently more steamtight than a multi-ring piston valve? The Nord Railway (France) reported‡:—

"Poppet valves and piston valves were tested out on the same engine and the latter showed themselves distinctly the better for the important reason that they are moving at the moment of uncovering and covering the ports. The wire-drawing is also reduced and the way the engine gains speed is quite remarkable.

"The piston valves have shown themselves to be tighter than the poppet valves."

The only test he himself had been able to make to confirm the views of the Nord Railway was to remove a front cylinder cover and, after placing the valve gear in mid-position, open the regulator fully. From his observations, the leakage past the valves was, if anything, in favour of the piston valve.

Again, the author's remarks about overcoming the inertia of the valve and valve gear, recalled views expressed by Wagner§.

"We have studied the poppet-valve problem, and I think that the best valve I have yet found is the one mentioned by Mr. Gresley, the Cossart valve, because that has not to be accelerated from a standstill in the very moment steam is admitted. The acceleration begins earlier than the steam admission. Take a system of movement that would make the whole Cossart or Kerkhove valve move in a way which would give the smallest possible acceleration, and therefore the slightest wear, and you have the link valve motion. So for this reason I have always stuck, so far, to piston valves and rather tried to improve their weak points."

It was significant that automobile engineers had often attempted to find something better than the poppet valve, and that any development in this direction had usually been towards sleeve valves, or cam shafts driven by coupling rods, in place of toothed gearing.

The Cossart and Franklin gears had link drives to the steam valves, and the valve had not to be moved suddenly from rest at

* Fry, L. H. 1927 Proc. I.Mech.E., p. 923, "Some Experimental Results from a Three Cylinder Compound Locomotive".

† Da Costa, G. 1939 Jl. Inst. Loco. Eng., vol. 29, p. 399, "The Indicator Diagram and the Efficiency of the Non-Condensing Simple Expansion Steam Locomotive".

‡ De Cuso 1933 Bulletin, International Railway Congress Assoc., vol. 15, p. 711, "New Suburban Locomotives for the French Nord Railway".

§ Wagner, R. P. 1935 Jl. Inst. Loco. Eng., vol. 25, p. 254, "High Speed and the Steam Locomotive".

admission. Trials conducted on the Great Indian Peninsula (G.I.P.) Railway had shown that the suddenness of admission with the Caprotti and rotary-cam Lentz gears induced steam-chest pressure oscillation.

At speeds of 300 r.p.m. and above, valve bounce could occur. This phenomenon was first suspected when indicator cards taken with the lever at 15 per cent cut-off were examined. It was noted that the diagram factor was higher than with the lever set at 20 or 25 per cent cut-off. It became apparent that due to valve bounce the actual cut-off was later than that shown by the lever reading. It seemed that the manufacturers would have to use two springs per valve of different natural frequencies—common practice in aeroplane and automobile engine design, to inhibit valve bounce at high speeds. Steam-loaded poppet valves were not free from this trouble and, furthermore, could give a lot of trouble due to valve stems sticking, even with the slightest trouble due to bad water conditions.

To date, all comparative trials published had been between Caprotti or Lentz rotary-cam gears and radial valve gears of 5- to 6½-inch valve travel. In the U.S.A. anything less than 7½-inch valve travel and 2-inch lap was not looked upon as a long-travel, long-lap gear, and a 9-inch valve travel was more representative of the modern trend.

It would be interesting to see comparative trials between a modern, cam-driven poppet valve gear and a modern, long-travel, long-lap radial valve gear fitted with multi-ring piston valves.

In the tests conducted in India with 4–6–0 type locomotives, Caprotti versus 5¾-inch travel piston-valves (D.4 class) and $6\frac{1}{16}$-inch travel piston-valves (D.5 class), the saving in fuel consumption in favour of the Caprotti locomotive was of the order of 4 per cent, but then only with cut-offs of about 15 per cent. For cut-offs of 25 per cent and longer there was no discernable difference. It was difficult to justify poppet valves on freight locomotives.

The rule of D. K. Clark regarding the volume of the steam-chest suggested that the G.I.P. Railway locomotives fitted with Caprotti gear had insufficient steam-chest volume. However, the tendency to-day was, he thought, to increase the diameter of piston valves unduly. For example, with 20½-inch diameter cylinders, long-travel piston valves of 10-inch diameter were ample, and fitting 12-inch diameter valves only threw an extra load on the valve gear, with no apparent gain. It seemed more desirable to increase steam-chest volume by increasing the diameter of the steam-chest (between the valve liners) to the fullest extent possible, rather than increase the diameter of the piston valves unduly. With long valve travel (not below 7½ inches) adequate port opening was assured.

In the author's equation, mean effective pressure was related to the speed of revolution and the value for the constant k was given for different types of engines. Data collected during trials on the G.I.P. Railway seemed to indicate that the constant k depended on the relation d^2/L, where d represented diameter of cylinder, and L, length of stroke.

Mr. J. N. COMPTON, M.I.Mech.E., wrote that the paper was of particular interest in India because analyses of valve-gear performance had developed there on very similar lines to those outlined.

In the earliest researches (1938), the basic diagram of comparison assumed release at the end of the stroke and a compression curve corresponding to the value giving theoretically the lowest steam rate in pounds per indicated horse-power-hour. More recent researches in India had been based on a diagram of comparison with a variable release taken as:—

Cut-off, per cent	10	20	30	40	50	60	70	80
Release	85	86	87	88	89	90	91	92

and the corresponding "ideal" compression curve was determined and used for completion of the card.

The laws of expansion and compression were taken respectively as $pv^{1\cdot2} = c$, and $pv^{1\cdot3} = c$, and analyses of indicator cards supported these polytropics rather than those accepted by the author.

The diagram factors of actual indicator cards, expressed as a fraction of these basic diagrams, were plotted versus piston speed, the comparison being made between engines of similar cylinder and wheel dimensions, but with different valve gears. The resultant diagrams were therefore comparable with the author's Fig. 20. Figs. 43 and 44 were for certain 4–6–0 and 2–8–0 locomotives with short-lap piston valves, increased-lap piston valves and for a type of poppet valve gear operated by rotating cams. The very marked superiority of the poppet valve gear at cut-offs earlier than 25 per cent was well brought out.

The dead-slow speed values did not give a 100 per cent diagram factor. This was explained by the fact that at these speeds the early release deprived the diagram of the fullness of its toe.

The analyses in India did not recognize the general relation $P_n = P_c\left(1-\frac{N}{k}^{1/x}\right)$ as acceptable; they favoured the introduction of additional factors—the operating cut-off, and the $\frac{d^2S}{D}$ ratio of the locomotive, where d and s represented diameter and stroke of cylinders, and D the diameter of driving wheels.

Regarding D. K. Clark's "forgotten" rule that the volume of the high-pressure steam chests should be equal to that of one cylinder, the fact that pressure drop from header to cylinder was a capacity effect, and was not due primarily to frictional resistance to steam flow, had been accepted in India for some time. In a recent specification for an engine with cylinders 20½ inches by 28 inches, the branch pipes were ordered as follows: "The minimum volume of each branch steam pipe between superheater header outlet and cylinder steam chest inlet shall be 2·5 cu. ft.", and the piston valve specified was 12 inches in diameter, L.M.S. type, so as to secure the maximum steam chest capacity.

Major-General A. E. DAVIDSON, C.B., D.S.O., M.I.Mech.E. (*Past-President*), wrote that curves in the paper appeared to assume that steam was used expansively *inside* the cylinder from the stated boiler (or possibly steam chest) pressure. In too many cases however (vide Fig. 12) this was far from being the case, and expansion occurred *outside* the cylinder. He asked what effect this had on the various curves. Contributors had referred to some of the causes which prevented full use being made of the maximum range of expansion from the full range of boiler pressure.

Throttling might take place at several points. The regulator might be used for throttling. The steam passages might not be designed, either as to size or geometry, so as to give the best aerodynamical flow. He drew attention to the care taken, in the case of the relatively small internal-combustion engines of motor-cycles, motor cars, and aero-engines, to plate and, or alternatively, polish the gas passages from and to the valves, after intensive experiment to secure the shape that gave the best streamlined passage. The valve design and sequence of valve events might limit the admission of steam to the cylinders, especially at high speeds. Sometimes the purchaser's specification stipulated certain limitations which unwittingly prevented the manufacturer from providing what his experience indicated to be the best conditions. Possibly some users placed a higher importance on certain practical running conditions than on efficiency in the cylinder. In this connexion, the principal railway companies were fortunate in that builder and user were part of the same organization.

If, for any of the reasons given above, the steam was not used, in the cylinders, as economically as it should be, further throttling effects might occur as a result of the boiler's inability to maintain the full supply of steam. Further, when the boiler was hard pressed its efficiency would fall from a figure of 75 or 80 per cent to 55 or 60 per cent, further aggravating the stoker's difficulties in maintaining the steam pressure. There might also be a throttling effect if the air passages through ashpan, tubes, etc., were inadequate for the higher rates of breathing.

In short, it was not from the cylinder alone that high efficiency resulted. Conditions and design outside the cylinder, and throughout the locomotive, might have a considerably greater

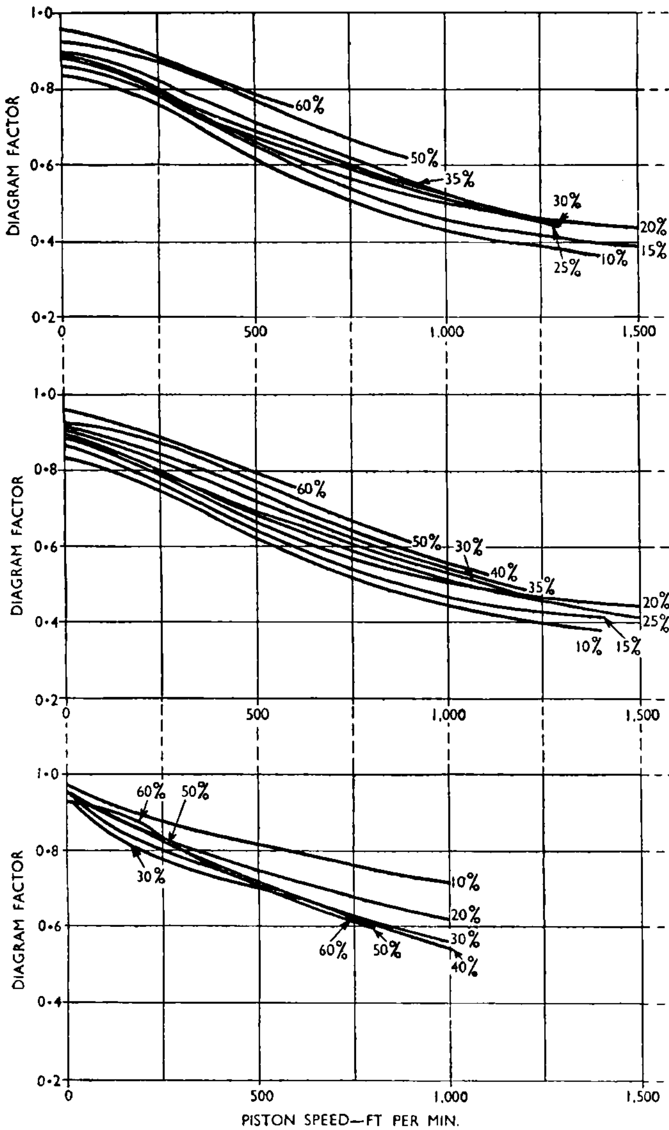


Fig. 43. Diagram Factors

4–6–0 locomotives having two cylinders 20½ inches by 26 inches, 10 per cent clearance, coupled wheel 6 ft. 2 ins. in diameter, and boiler working pressure 180 lb. per sq. in.

(1) Walschaert gear piston valve 1½-inch steam lap.
(2) Walschaert gear piston valve 1¾-inch steam lap.
(3) Poppet valve gear operated by rotating type cam.

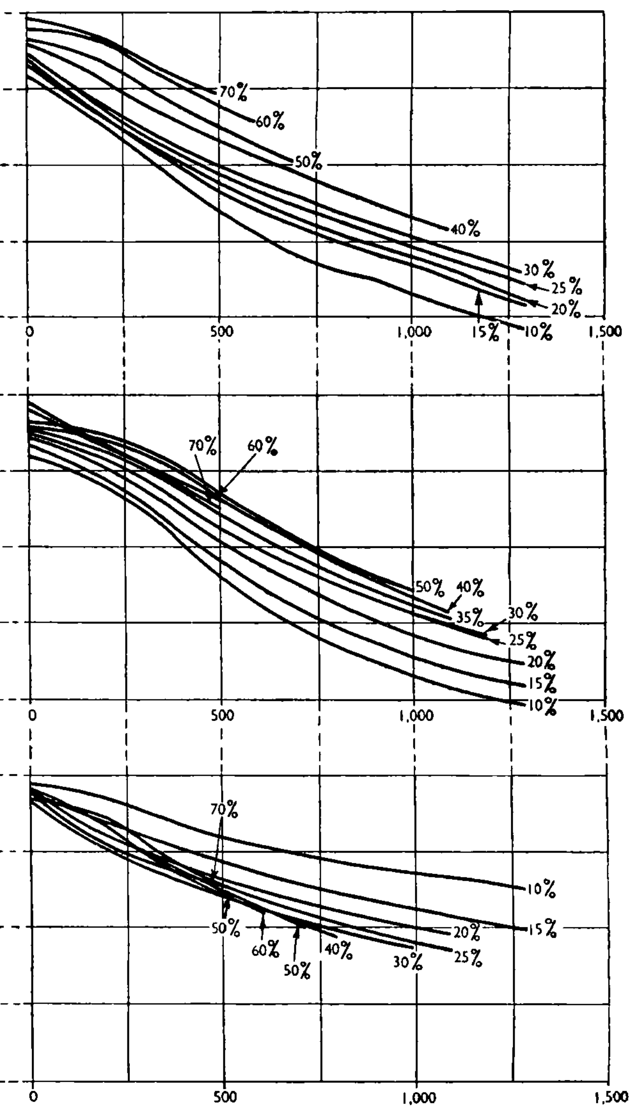


Fig. 44. Diagram Factors

2–8–0 locomotives having two cylinders 22 inches by 26 inches, 10 per cent clearance, coupled wheel 4 ft. 8½ ins. in diameter, and boiler working pressure 180 lb. per sq. in.

(1) Walschaert gear piston valve 1-inch steam lap.
(2) Walschaert gear piston valve 1¼-inch steam lap.
(3) Poppet valve gear operated by rotating type cam.

bearing on the attainment of the desired power output. The complete locomotive was an entity; efficiency was required in all its parts, if full output had to be attained and maintained.

Mr. P. C. Dewhurst, M.I.Mech.E., agreed with the author on the importance of factors connected with the actual utilization of the steam in order to obtain optimum results, although he himself considered that fully adequate boilers were the most indispensable of all things in effective locomotive operation.

He strongly supported the author's recommendation that, when all safeguards had been provided in assessing the amount of steam required of the boiler, by the engine, this amount be increased by at least 25 per cent to cover the various "imponderables" of railway locomotive operation. He had never known a locomotive boiler "too big" for the engine, whereas between 60 per cent and 80 per cent of the original boilers on the various railways whose locomotives had been in his charge were "too small". Even now, it was sometimes erroneously supposed that a larger boiler than strictly necessary was extravagant in fuel, whereas such a boiler merely operated at an output ratio more favourable to efficient combustion. He recollected a particular group of locomotives which, when given boilers of nearly 60 per cent increased capacity, burnt 20 per cent less fuel than with the original boilers—even per unit of ton-mile—when employed on trains no heavier than those hauled previously. From the operating viewpoint some features less than ideal could be "carried", but no excellence of these would nullify the fault of an insufficient boiler capacity.

An appreciable proportion of the author's exposition resolved itself into an exceedingly strong—irrefutable—case, which it would be hard to better, for a modern design (i.e. using 275–350 lb. per sq. in. boiler pressure) of compound locomotive, and this was, in his own opinion quite correctly, explicitly stated in various places in the paper. He felt that the matter of compounding of locomotives was most ripe for reinvestigation in the light of the altered circumstances—both technical and economic—of the present day.

Mr. P. L. Falconer (Argentina) wrote that in his detailed description of the developments that had taken place in the motive portion of the steam locomotives, the author had focused attention on those outstanding developments.

Formulae based on those improvements were very useful and necessary, but to take those alone, and to class others, based on boiler characteristics, as fundamentally wrong, might be good English but bad American.

In American practice the economies of burning low grade fuels and the emphasis on the transport capacity rather than the thermal efficiency alone, had led to designs in which the boiler was the backbone of the design. Their engine design did not suffer to anything like the extent that European design did from the German school of thought which, in introducing superheating, advocated smaller valves and lower boiler pressures.

The development of the locomotive power at speed depended on the use of new developments in conjunction with, rather than as alternatives to, established improvements.

Boiler pressure, apart from helping the power-weight ratio, like superheating permitted the generation of steam with a lower latent-heat/total-heat ratio, a feature of more importance in non-condensing engines than generally recognized.

Compound expansion recovered much of the steam leaking past the low-pressure piston valves, and heat lost due to the passage of high temperature steam at high velocity through ports cooled by the exhaust steam. Mean cylinder temperatures were much higher in the high-pressure cylinder of a compound engine than in a simple engine, and cylinder lubrication trouble increased.

The effects of restricted steam flow were much greater in the compound. In 1925 he had published details of indicator trials carried out in compound engines, showing the advantage of improved port areas on the exhaust side of the high-pressure cylinder, and used 14-inch diameter valves on a 21-inch cylinder on locomotives introduced at that date*.

The need for greater port area on the exhaust side was not at that date fully appreciated. The French compounds then used an excessive high-pressure cylinder clearance to overcome the resulting compression. The Baldwin three-cylinder, high-pressure locomotive and the four-cylinder, high-pressure compound introduced by Gresley on the L.N.E.R. both suffered from this defect, and the matter was finally cleared up by the classic work of M. Chapelon in the later French compounds.

The author had emphasized the advantages of poppet valves. The writer, being associated with their introduction and maintenance on main-line passenger engines introduced in an Argentine railway, would endorse his findings, but poppet valves fitted to obsolete locomotive designs might be disappointing.

In modern locomotive practice there were at least two further features that should be mentioned—better use of adhesive weight and the control of the surface tension of the boiler water.

Although Gresley's early Pacific and his compound design suffered badly through poor port design, he used the ample boiler feature of the earlier Atlantic design and focused attention on the better use of adhesive weight which later revolutionized American locomotive design.

Many locomotive designs had suffered from the fact that they were only cylindered to take advantage of the adhesion at starting or on very steep grades; on the remainder of the journey less than 50 per cent of this adhesive weight was used. The provision of ample cylinder power to complement the generous boiler power took better advantage of the adhesive weight at high speed and a fuller use of the characteristic advantage of poppet valves.

By making use of these features and the even torque of the three-cylinder design, he had found it possible to double the horse-power with an increase of only 17 per cent in adhesive weight and 25·7 per cent increase in total engine weight.

With the larger boilers, the firebox crown was much higher in the boiler, and steam space was less than in earlier designs. Most boiler waters contain minerals that tended to alter the surface tension of the water, and at high rates of evaporation the water level was raised and foaming and "carry over" might result.

The fine clearance of the water-packed stems of the poppet valves were sensitive to the effects of "carry over".

For the development of power at speed it was essential for the surface tension of the boiler water to be kept under careful control.

* Jl. Inst. Locomotive Eng.

Mr. Lawford H. Fry, M.I.Mech.E., wrote that the paper brought very clearly to light the effect that variations in boiler pressure, clearance, cut-off and compression had on the performance of locomotive cylinders.

The two parts of the paper would be discussed separately. The author's basic indicator-cards served usefully to show the effect of boiler pressure and cut-off when the admission steam temperature and the back pressure were uniform, and they closely resembled the idealized cards that he himself put forward twenty-one years ago under the name of the "locomotive cycle"†. At that time the author did not take kindly to the type of card proposed; but expressed the opinion that it was proposed "to hide the inefficiency of the old steam locomotive". He was glad that the author's view had softened over the years (compare the author's Fig. 12 with his own Fig. 8). The only difference was that he himself had used actual admission pressures and temperatures, and the measured exhaust pressures, while the author had assumed steam admitted at boiler pressure, and assumed a uniform temperature of 600 deg. F. and an exhaust pressure of 18 lb. per sq. in. abs. for all conditions.

The author's Figs. 1 and 2 showed the mean effective pressure calculated from his basic cards for a wide range of pressures, cut-offs and clearance values. It might be of interest to note that, while the author had computed his cards on the assumption that expansion and compression followed the pv = constant law, he himself had checked a number of cards using an enthalpy–entropy chart with specific volume curves added; very close agreement was obtained.

In the series of curves showing the specific steam consumption per indicated horse-power-hour as affected by steam pressure, clearance, and cut-off, it seemed curious that the best steam rates were obtained with pressures of 230 lb. per sq. in. and that higher pressures gave poorer steam rates. The author pointed out that the Rankine efficiency increased with the boiler pressure, but that the effect of clearance neutralized this gain. He thought that the problem deserved further discussion. If high steam pressures were used to obtain increased power with limited space and weight, it was highly desirable to utilize at least part of the greater Rankine efficiency; to do this a high ratio of expansion was necessary. Low clearance-values were required as shown by Fig. 6 of the paper, but it should also be pointed out that a high degree of compression produced the same effect as a low clearance volume. If m and c were respectively compression and clearance percentages of cylinder volume, and v_1 and v_3 were specific volumes at admission and at the beginning of compression, it could be shown that the steam to be supplied per stroke would be a minimum when $m = c(v_3 - v_1)/v_1$. If v_3 remained constant, an increase in admission pressure would reduce the specific volume v_1 and would increase the ratio $(v_3 - v_1)/v_1$. Therefore to maintain a minimum value for the weight of steam to be supplied, the compression m must be increased or the clearance c must be reduced. With high steam pressures, cam-operated poppet valves had the advantage of permitting the compression to be set independently of other valve events.

There was less certainty about the computed steam rates than the computed mean effective pressures. With given conditions of expansion, it was possible to lay down an ideal indicator card and to compute its area quite definitely. This established the work produced per stroke, but the weight of steam required to do this work was a more elusive quantity. It depended on the weight of steam trapped for compression. The volume of this steam was known, but its specific volume was a matter of some uncertainty. The author apparently assumed that the quality of the steam as compression began was that which would be obtained by isentropic expansion from admission pressure to the exhaust pressure. He preferred to assume isentropic expansion followed by isothermal release: at release the steam did no mechanical work, and loss of heat to the cylinder could not be great; as a matter of fact the cylinder walls at the end of the stroke were hotter than the steam. He considered the expansion

† Fry, L. H. 1927 Proc. I.Mech.E., p. 923, "Some Experimental Results from a Three-Cylinder Compound Locomotive".

of steam at 185 lb. per sq. in. gauge and 600 deg. F. at admission, with 10 per cent clearance and 25 per cent cut-off. The steam entering with specific volume of 3·0 cu. ft. per lb., entropy of 1·676 and enthalpy of 1,321 B.Th.U. per lb., would expand isentropically 3·14 times, to have at the end of the stroke a release pressure of 44 lb. per sq. in. abs. and enthalpy of 1,178 B.Th.U. per lb. At release the pressure was to drop to 18 lb. per sq. in. With isothermal release the quality of the exhaust steam would be determined by its pressure and its enthalpy—of 1,178 B.Th.U. per lb.; the steam would have a specific volume of 24 cu. ft. per lb. and about 50 deg. F. superheat. On the author's assumption, the exhaust steam would have its original entropy of 1·676 at the pressure of 18 lb. per sq. in. abs.; this would correspond to a specific volume of 20·4 and enthalpy of 1,110, and the steam would be about 4·6 per cent wet. With the higher specific volume due to isothermal release, the weight of steam compressed would be about 18 per cent less than with an assumed isentropic expansion down to the release pressure, and there would be a corresponding increase in the amount of steam to be supplied per stroke from the boiler.

TABLE 9. COMPUTED STEAM RATES

Boiler pressure, lb. per sq. in.	135			185			285		
Cut-off, per cent	10	25	40	10	25	40	10	25	40
Steam rates, lb. per i.h.p.-hr.:									
At 600 deg. F.	14·1	15·3	17·4	13·2	14·3	16·7	13·1	14·7	16·2
At 700 deg. F.	13·1	13·6	15·2	11·6	13·1	14·5	11·5	12·8	13·9
Reduction for 700 deg. F., per cent	7·1	11·1	13·4	12·1	8·4	12·6	12·2	12·9	14·2

TABLE 10. P.R.R. E-6S LOCOMOTIVE NO. 51, BULLETIN NO. 27

Test No.	Test Designation	Speed, r.p.m.	Cut-off, per cent	Mean effective pressure, lb. per sq. in.
3854	120–15–F	120	16·8	46
52	120–25–F	120	27·1	70
47	120–35–F	120	38·1	94
53	120–45–F	120	42·4	110
51	120–45–F	120	43·4	109
3855	160–15–F	160	17·9	44
33	160–25–F	160	25·3	58
35	160–35–F	160	41·1	90
50	160–45–F	160	44·5	103
49	160–50–F	160	49·8	114
3836	200–15–F	200	20·1	42
37	200–25–F	200	28·8	61
07	200–35–F	200	37·7	85
09	200–45–F	200	44·5	96
48	200–50–F	200	52·0	104
3832	240–15–F	240	20·1	38
44	240–25–F	240	29·6	56
31	240–35–F	240	39·7	79
10	240–45–F	240	47·7	86
3845	280–15–F	280	17·9	38
08	280–25–F	280	24·8	54
42	280–35–F	280	43·1	71
11	280–40–F	280	45·2	76
3846	320–15–F	320	18·2	35
34	320–25–F	320	30·2	44
38	320–35–F	320	41·5	66
3841	360–15–F	360	19·0	33
40	360–25–F	360	30·9	48
39	360–30–F	360	34·9	54
43	360–35–F	360	42·1	56

Clearance 10·4 per cent.

The assumption of isothermal release gave an exhaust steam condition more nearly that obtained in practice and had the advantage of giving a more conservative estimate of the steam required per i.h.p.-hr. The author's steam rates would be increased by about 5 per cent if computed on the basis of isothermal release.

The author based all of his computations on an admission temperature of 600 deg. F. Table 9 showed the increase in efficiency obtainable with 700 deg. F. Computation for both temperatures was based on isothermal release.

TABLE 11. P.R.R. K-2SA LOCOMOTIVE NO. 877, BULLETIN NO. 18

Test No.	Test Designation	Speed, r.p.m.	Cut-off, per cent	Mean effective pressure, lb. per sq. in.
3008	120–20–F	120	20·1	67
1	120–25–F	120	21·4	70
2	120–30–F	120	28·4	81
3	120–40–F	120	39·6	104
4	120–40–F	120	40·6	105
3005	160–25–F	160	23·3	67
6	160–30–F	160	30·2	78
7	160–25–F	160	35·2	87
9	160–35–F	160	33·4	90
27	160–45–F	160	44·9	107
29	160–50–F	160	49·3	110
3010	200–20–F	200	22·7	62
28	200–25–F	200	29·6	73
11	200–35–F	200	33·8	82
16	200–50–F	200	50·0	99
17	200–50–F	200	50·9	101
3023	240–20–F	240	24·8	59
12	240–25–F	240	28·0	69
21	240–30–F	240	33·9	74
13	240–35–F	240	36·3	78
20	240–50–F	240	50·4	86
3025	280–20–F	280	24·8	56
14	280–25–F	280	29·8	71
26	280–30–F	280	35·4	69
15	280–35–F	280	35·7	74
3024	320–20–F	320	23·5	55
19	320–25–F	320	30·6	62
22	320–30–F	320	35·1	66
3030	360–30–F	360	38·5	62

Clearance 13·4 per cent.

The advantage of using high superheat, particularly with high pressures and long cut-offs, was obvious.

The mean effective pressure was not affected by the change in superheat, but the quantity of steam required per stroke, for given cylinder conditions, was smaller for the higher temperature.

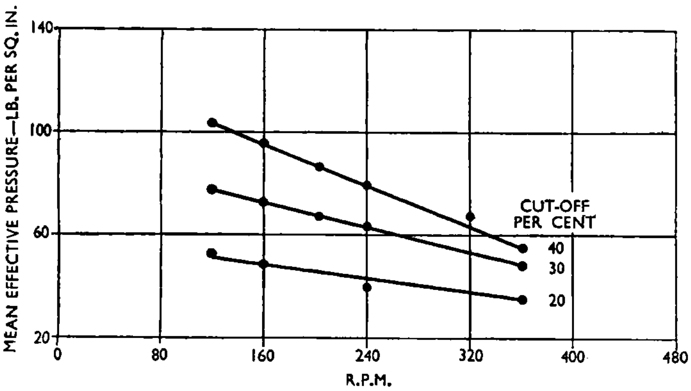
The author had provided a most interesting method for the development of characteristic power equations for comparative evaluation of locomotives for which test results were available. The high rating given to the Pennsylvania Railroad T-1 locomotive was noteworthy.

Regarding details of the method, he noted that the symbol k adopted for the characteristic power equations differed from the one first proposed in the more general form of the equation. In Figs. 14–19, each cut-off curve had an individual value of k; what relation did this bear to the mean value k in the characteristic power equation? The mean value, for any locomotive,

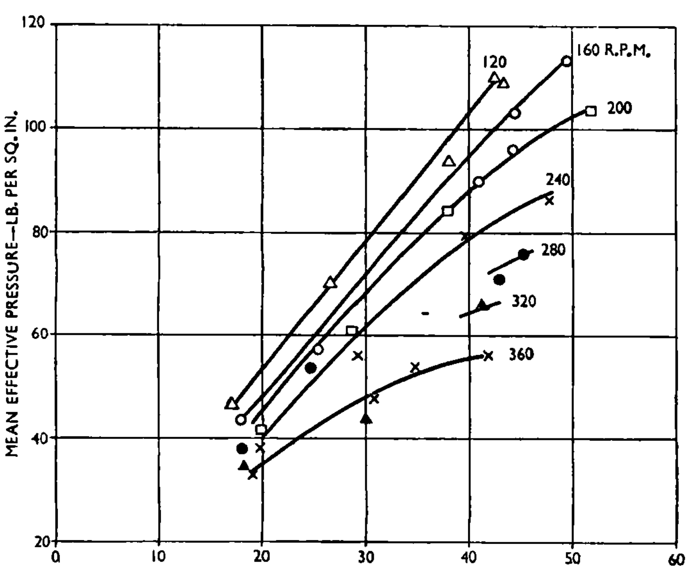
TABLE 12. P.R.R. T-1 LOCOMOTIVE No. 6110

Test No.	Test Designation	Speed, r.p.m.	Cut-off, per cent	Mean effective pressure, lb. per sq. in.
1441	160–10–F	160	11	78
30	160–15–F	160	18	110
15	160–25–F	160	25	138
1440	240–10–F	240	10	74
37	240–25–F	240	25	129
1436	280–15–F	280	15	98
1419	320–10–F	320	11	73
32	320–20–F	320	23	113
11	320–25–F	320	27	118
1412	360–15–F	360	22	95
38	360–20–F	360	28	110
14	360–25–F	360	30	116
1444	400–20–F	400	30	114

Clearance 13·5 per cent.



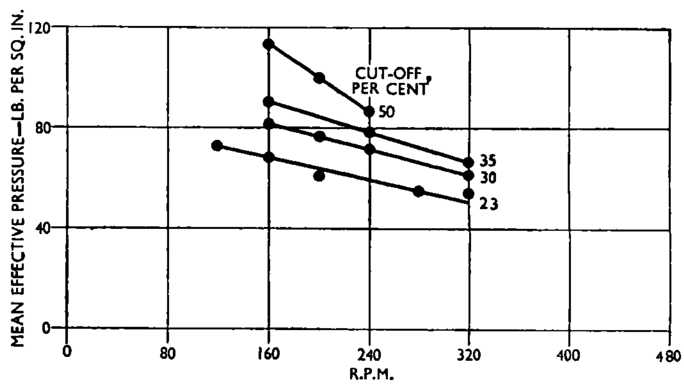
b



a

Fig. 45. Mean Effective Pressure versus Rotational Speed and Cut-off, for Pennsylvania Railroad Locomotive E-6s, No. 51

seemed to be rather arbitrary: it would depend very largely on the number of different cut-offs at which the locomotive had been tested. There was no systematic relation between the length of the cut-off and the value of k: in Fig. 15, for example, the long cut-off carried a high value of k, while in Fig. 16 the reverse was true.



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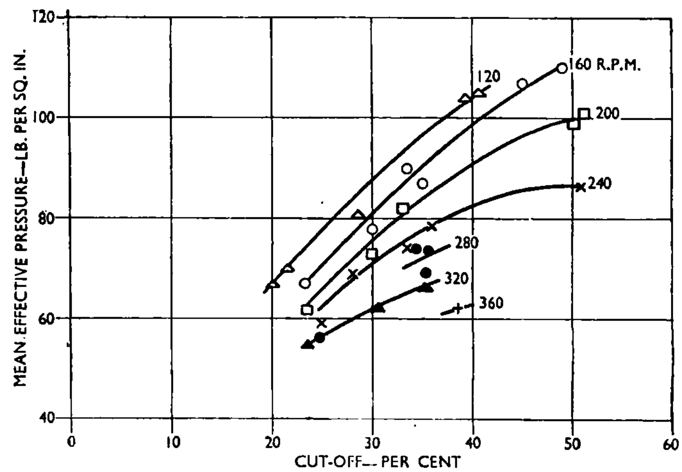

a

Fig. 46. Mean Effective Pressure versus Rotational Speed and Cut-off, for Pennsylvania Railroad Locomotive K-2sa, No. 877

He had plotted again the basic data for the Pennsylvania Railroad locomotives represented in Figs. 16, 17, and 19. For the E-6s and the K-2sa locomotives, Tables 10 and 11 covered all of the tests reported by the Pennsylvania Railroad—30 and 29 in number respectively, whereas the author had plotted 21 tests in each case. In the case of the T-1 locomotive, Table 12 included the 13 tests used by the author in Fig. 19, but the points plotted in Fig. 47 represented all of the 49 tests made at Altoona; the complete report on the T-1 had not been published.

There were certain differences between his Figs. 45, 46, and 47 and the author's Figs. 16, 17, and 19: the author had apparently assumed that the test designations corresponded exactly to the cut-offs at which the tests were run, whereas, in practice, the speed was held very exactly to the nominal value, but the cut-off often varied considerably from that prescribed. As shown in Tables 10, 11, and 12, each test was given a designation—such as 160–25–F, which indicated that the test was nominally to be run at 160 r.p.m. with 25 per cent cut-off and full throttle. In Table 12 it would be seen that the T-1 locomotive in the nominal 20 per cent cut-off series had actual cut-offs ranging between 23 and 30 per cent: the author's direct method of plotting was not reliable. To find the effect of speed on mean effective pressure with a constant value of cut-off, it was necessary first to plot mean effective pressure against cut-off. When this was done as in Figs. 45*a*, 46*a*, and 47*a*, it was seen that the points grouped themselves according to speeds. Having drawn smooth curves for each speed it was possible to plot, as in Figs. 45*b*, 46*b*, and 47*b*, mean effective pressure against speed for any selected percentage of cut-off. Examination of the whole body of the tests led him to the conclusion that drop in mean effective pressure was not proved proportional to the square root of the speed. In Figs. 45*b* and 46*b*, for the E-6s

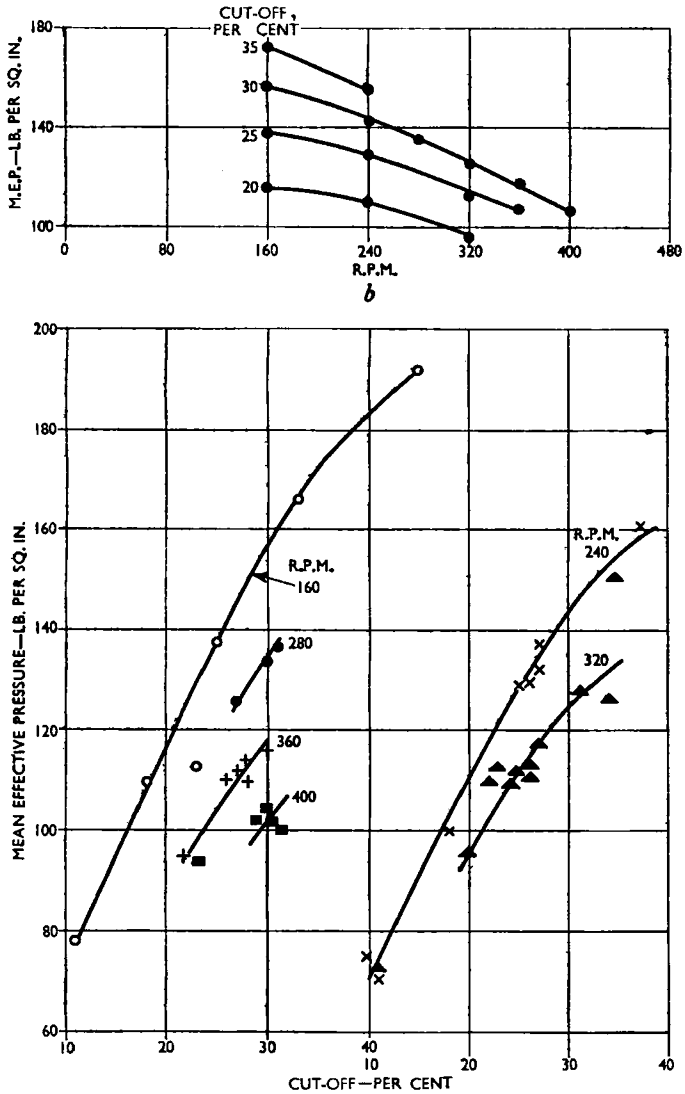

Fig. 47. Mean Effective Pressure versus Rotational Speed and Cut-off, for Pennsylvania Railroad Locomotive T-1, No. 6110

and K-2sa locomotives there was a definite straight line relation between mean effective pressure and speed at all speeds between 120 and 280 r.p.m., with a slight uncertainty about the values at 320 and 360 r.p.m. Fig. 47*b*, for the T-1 locomotive, showed a slight downward concavity. The square root relation would require an upward concavity.

Mr. F. Mills (Perth, W. Australia) wrote that the demonstration, that the relationship between the mean effective pressures of an actual indicator diagram and an assumed ideal diagram could be expressed as an exponential law, was of the first importance, since it not only afforded a sound basis for general comparison of performances, but formed an excellent tool for analysis of test results.

Presumably the author, when referring to the effect of increased back pressure in promoting relative cylinder efficiency, was dealing with ideal diagrams showing the back pressure actually existing. If comparisons were made in all cases with an ideal diagram showing 18 lb. per sq. in. back-pressure, then cylinder efficiency should fall as back pressure rose. It was difficult to assess this effect without further research, since high back pressure might result in compression to admission pressure and in this way the evil of high back pressure might serve to counteract the effect of large clearance in some measure.

Some relationship of this sort appeared to exist in the remarkable Pennyslvania T-1 locomotive, which must have large clearance with its large valves and steam passages. The designer had adopted four simple cylinders instead of two, mainly to improve steam distribution, despite the increase of clearance entailed. Further the blast of this engine had been sharpened to ensure adequate steam production and it was understood the back pressure was far above the assumed ideal.

The contention that compounding appeared essential to high efficiency with pressures exceeding 250 lb. per sq. in. did not gain strong support from the test data given. Of all the examples cited the best was the Pennsylvania T-1 with 300 lb. per sq. in. and simple expansion. Even the Paris-Orleans No. 3705 locomotive was far behind, and the author's comment that this engine might have been the best "if the clearance volume could have been kept to a normal value with the enlarged poppet valves and steam passages" was illuminating. Large clearance was physically inseparable from large valves and steam passages and, though a further design of the compound might show an improvement, there was no evidence of such a margin of superiority over simple expansion engines as to warrant the complication and difficulties of compounding. On the evidence so painstakingly compiled the limit of pressure for simple expansion had not yet been reached.

Fresh emphasis was given by this research to the need to concentrate effort upon improvement of admission and realization of high superheat. In Fig. 13 it was shown that piston valves of special design were nearly equal to the best poppet valves tried, but the severe curtailment of steam passage area, caused by reduction of cut-off with all kinds of valve, was thrown into sharp relief. Usually only the maximum valve-travel was quoted, but this was used only at very low speeds; actual travel at speed was very much less and inertia forces were correspondingly less. This suggested that much longer travel could be used, provided full gear was not applied when drifting at high speeds.

Since the main objective was increase of steam-passage area, there was good reason to explore the practicability of enlarging valve diameter. For example, the rebuilt Royal Scot would have given a better card than Fig. 12 if the piston valves had been made 10 inches or 12 inches in diameter. This would have entailed appreciable increase of clearance, but that was a secondary consideration.

Any attempt to standardize ratios of proportions of locomotive parts was to be deplored, but the author's investigations suggested the possibility of some general minima. It appeared that piston valves should not be less than half the piston diameter, and a minimum area of steam circuit in relation to piston area seemed desirable. The suggested value $k = 45$ for "long-lap" piston valves appeared appropriate to the Royal Scot design, but in itself suggested considerable room for improvement of piston-valve designs.

Mr. T. Robson (Darlington) agreed with the author that the usual clearance volumes and ratio of expansion at present possible in one cylinder made the use of increasing boiler pressures uneconomical without compound expansion: they had not yet succeeded in making the best possible use, in a single-expansion locomotive, of steam pressures and temperatures that had been in use for twenty years.

Many engineers had drawn attention to this; he himself had submitted, ten years ago, original experimental data which he hoped would prove convincing. There was a long time-lag before new ideas were generally accepted—e.g. the long-lap valve, the most important and most recent economic improvement in the single-expansion locomotive since superheating, was brought out by Churchward about forty years ago, but twenty years had to elapse before it was generally adopted.

There had been great advances made in the development of power at speed in France and America. There had been nothing comparable in Great Britain, because the large engines that were built shortly after the main line railways were amalgamated into four large groups were found adequate to deal with the loads and speeds required. The economy in coal consumption, due to superheating and the eventual adoption of the long-lap valve, was so great when compared with the very poor results from the older engines that it was considered satisfactory, having regard to the many more pressing administrative problems.

The lower diagram in Fig. 12 was not representative of early cut-off working with Walschaert's gear, because the lead of $\frac{5}{16}$ inch was about twice as much as usual, resulting in a much better steam-port opening and therefore a higher value for k than was possible with a normal amount of lead. Although the diagram was as good as could be expected from a piston valve, the large clearance-volume considerably limited the economical use of the high steam pressure.

He wished to stress that, in using the author's method of finding k, it was necessary to compare the actual cut-off with the graduations on the reversing gear, particularly in early cut-off positions, and to correct these: a very slight movement of the reversing gear in the early cut-off positions changed the mean effective pressure, and the drawbar pull, very considerably.

Removing all other variables, by making constant-speed tests on the level using a counter-pressure brake-engine, had shown that the drawbar pull of a modern type locomotive in 15 per cent cut-off was more than doubled at ordinary running speeds by moving the reversing wheel half a turn towards full gear.

The writer had been present during tests with the latest Chapelon compounds and was impressed with the large drawbar-pull at high speed and early cut-off. Expressed as a percentage of the rated tractive effort, it was at least twice as great as that possible with conventional single-expansion locomotives, so it was surprising that the value of k shown in Fig. 15 was not higher.

The low value of k shown in Fig. 14 for the original locomotive might be due to condensation in the low-pressure cylinders owing to insufficient superheat. In the later engines this was increased; he had noted inlet-steam temperatures as high as 800 deg. F. Probably these compounds were the first that had had dry steam at the low-pressure exhaust; this, in his opinion, was a very important reason for their success. Moreover, as a result of two-stage expansion, there was less port restriction, when working in an economical cut-off at high speed, than with single expansion.

For the provision, with a single-expansion locomotive, of greater power at high speed, the obvious devices of building larger locomotives, or working existing ones in longer and therefore less economical cut-off positions, or using larger piston valves, were presumably considered too wasteful; the aim was to obtain a much higher proportion of the rated tractive effort than at present, when working in an early cut-off. It could not be done by a piston valve: natural laws governing steam flow precluded such a possibility. It could only be done by using independent steam and exhaust valves having wide and straight port openings and passages.

The slope of the curves for the poppet-valve engines on Figs. 18 and 19 was very good, but the values of k for the T-1 locomotive varied a good deal—perhaps as a result of the inaccuracy in graduation already referred to. The flatness of these curves for the different cut-off positions could be attributed in part to the large steam openings, and in part to the valve springs. Above a certain speed the cut-off started to lengthen without alteration in the reversing gear, owing to the increasing kinetic energy in the valve and the increasing force necessary to close it in the decreasing time. There could also be peculiar surge effects in the springs

At about 75 m.p.h., with a usual size of driving wheel, the time interval for steam admission in 15 per cent cut-off was $\frac{1}{40}$-second, but a $\frac{1}{100}$-second time lag in closing would increase the cut-off to 25 per cent. The delayed closing of the exhaust valve increased the amount of live steam required to fill the clearance space. The combined effect gave a higher mean effective pressure than could be obtained if the valve events were true as with a positively closed piston valve, and a false value of k at high speeds, making the valve appear better than it really was.

The writer had made many tests with poppet-valve locomotives, including comparative tests between the rotary-cam type and long-lap piston valves, modern passenger engines being available that were identical in all respects except the valve gear. These tests were made at a series of constant speeds in different reversing-gear positions using a counter-pressure brake engine.

The results from two poppet-valve engines were practically identical, but were disappointing when compared with the piston valve, which showed much greater economy over a wide range of power at ordinary running speeds. These engines had four valves per cylinder; the effect explained would be less with eight valves per cylinder but could not be eliminated. In addition to this, the sudden changes in direction through the poppet valve and the steam passages were just as bad as with the piston valve, and hindered the free steam-flow during the short time interval; also the clearance volume was too large. In his opinion, the performance of the poppet valve at high speeds was so far from the ideal, and eight valves per cylinder were such a complication, that it would be better to start developing a valve of new type free from these defects.

As the time interval of admission was $\frac{1}{40}$-second or less, to get the maximum possible weight of steam into the cylinder and the minimum pressure drop (from boiler pressure) at cut-off, a type of valve giving a wide opening to a straight steam-port of large area, through a passage as short as possible to keep the clearance volume small, and leading directly into the cylinder without change of direction, was required. There was no other way of getting greater power at high speed with high thermal efficiency. In his opinion, direct openings of adequate size could be provided only by the use of rotating valves for admission and exhaust.

The author compared the relative usefulness of a locomotive testing plant and a single-cylinder unit, for making improvements in the steam circuit. The single-cylinder unit was more useful, because alterations could be made more cheaply, and it was, in his own opinion, the key feature in design. If economic development was required, such a unit must be provided, and engineers with the necessary aptitude must be concentrated on the problem.

A unit built up in sections, to enable alteration to be made cheaply, was suggested in this country thirteen years ago. It was a mistake to think that locomotive development in this country had been delayed owing to the lack of a testing plant: it was due to the concentration at the grouping of so much responsibility on so few chief engineers. Whilst great improvements were made in workshop methods, administrative work and correspondence left insufficient time for the problems met with in development work. A lot of experimental work had been done before the war, and this showed where improvements were desirable; it remained only to make a start.

What a difference it would make if those directing large undertakings planned a less concentrated organization, in which experienced engineers with inventive ability were free to give of their best; he feared nationalization would put an end to such a hope, and this failure to harness the individuals creative instinct was the primary handicap to development.

Mr. D. W. SANFORD (Derby) considered that higher pressure was only advantageous in increasing the efficiency if an adequate expansion ratio allowed it to be fully utilized, and this seemed to require the use of compounding. Efficiency might be sacrified, of course, in favour of a better power/weight ratio. The author mentioned the effect of leakage which might be of considerable importance unless piston valves were carefully designed and maintained.

He was not sure that the author was right in discarding the old idea that the power of a locomotive depended upon the power of the boiler. He was quite correct in saying that some of the older engines could make more steam than they could usefully employ, and would steam freely however hard they were worked, but that, he suggested, was due to the inefficiency of their cylinders. The greater the quantity of heat which was turned into useful work in the cylinders, the less was the energy left in the exhaust to produce the necessary draught on the fire. Consequently, when the cylinder efficiency improved and more energy was taken out of the steam, it became necessary to increase the boiler capacity or to increase the efficiency of the smokebox arrangement in order to maintain steam.

If one were losing time with a train (because insufficient work was being done in the cylinders) but had plenty of steam, one could lengthen the cut-off, but if time was being lost as a result of poor steaming, this could be remedied, if at all, only by better work on the part of the fireman. Thus he believed that the power of the engine still depended on the amount of steam which could be produced in the boiler.

The author's method of predicting the mean effective pressure at any speed or cut-off would no doubt enable the designer to fix the cylinder volume required to produce the necessary power at that speed and cut-off, but he himself thought the technical staff of any of the larger railways already had fairly good information on that point from the indicator cards taken from existing engines. They could give a fairly good estimate of the power that would be developed at any speed and at any selected cut-off for any new design they were producing. They did not know, with anything like the same degree of certainty, the best cut-off to use at any given speed.

There was every reason to suppose (as indeed the author had shown in Fig. 6) that the efficiency was dependent on the cut-off and the speed, and the maximum efficiency did not necessarily coincide with the earliest cut-off. It was to be expected that at low speeds, where leakage and heat transference to the cylinder walls had more time to operate, the cut-off for maximum efficiency would be somewhat later than would be the case at higher speeds.

The point of maximum efficiency was governed by so many factors that it could not be calculated; it could be obtained only by constant-speed tests either on the road or, preferably, on a test plant.

If the designer had this information, for the particular boiler pressure and superheat (which had probably already been selected on practical considerations), and if he knew the train loads, gradients, and speeds required therewith, he could determine the cylinder volume giving greatest efficiency under normal conditions.

The real commercial efficiency of a locomotive was represented by the pounds of coal (or the number of thermal units) supplied to the firebox per drawbar horse-power-hour. Considerations of efficiency in terms of indicated horse-power were misleading: a very large and heavy locomotive might burn more coal per drawbar horse-power-hour than a lighter engine of lower thermal efficiency. A locomotive had to produce power at the drawbar at a minimum cost; the power expended in moving the engine itself was of no value.

Dr. W. A. Tuplin, M.I.Mech.E., wrote that the author's relation between mean effective pressure and speed was not strictly an "exponential" one: that description applied to an expression in which the independent variable appeared as an exponent, or index, of some constant. This comment was made in order to draw attention to the fact that the mean effective pressure fell to zero, in his opinion, only at infinite speed, and whilst an exponential law could be drawn up to agree with this, the author's expression (like that of Dalby) did not do so; it suggested zero mean effective pressure at a finite speed.

The author stated that "boiler pressures have been increased primarily in order to increase the power obtainable within the limits of the prescribed loading gauge and axle weight". This might be very seriously questioned because

(*a*) higher pressure did not increase the output of a boiler of specified major dimensions,

(*b*) higher pressure compelled increase in the weight of a boiler of specified major dimensions,

(*c*) higher pressure did not in itself appreciably raise the efficiency with which the steam was used.

Higher pressure could help in the direction mentioned by the author only by permitting the development of greater power in cylinders of given size, but this was hardly a practical point as there were at present comparatively few British locomotives whose cylinders were as large as they might be. In other words, nearly every design might be modified to work equally well at lower boiler pressure.

Increase in boiler pressure *reduced* the power/weight ratio of the locomotive, as the saving in weight in the cylinders was small compared with the increase in weight of the boiler.

He suggested that it was preferable in a discussion of this sort to specify "expansion ratio" instead of "cut off"; then the thermodynamic effect of clearance was automatically taken into account and need not be mentioned.

As steam trapped in the cylinder at the compression point was not lost, variation in clearance volume could not effect efficiency for any given expansion ratio, although it did influence the possible power output and the piston loading in the approach to dead centre.

The advantages of compounding were:—

(*a*) smaller temperature range in each cylinder for any given overall expansion ratio,

(*b*) reduced difficulty with compression pressures because of the later cut-off (and therefore later compression point) in each cylinder for any given overall expansion ratio.

The use of independent inlet and exhaust valves in a single-expansion locomotive nullified (*b*) and did much towards nullifying (*a*) because the valves and ports (where the steam came most intimately into contact with metal) were maintained at constant temperatures.

The advantages of compounding appeared to be almost independent of boiler pressure, but if a compound were to compete with a simple that could be built within the same space limitations, the compound would require a boiler pressure nearly twice as high as the lowest that would suffice for the simple, because the degree to which the steam could be expanded was determined by the volume of the low-pressure cylinders, and this was for obvious practical reasons, only about half the possible cylinder volume for a four-cylinder simple.

The maximum speed at which the full boiler output of steam could be economically used was largely determined by the ratio of area of inlet port opening to the volume of steam in the clearance and the cylinder at the point of cut-off. That maximum speed could be increased by increasing the lap of the valve and consequently its travel and the port opening at any specified cut-off. Appropriate design of the valve gear could be used to combat increased inertia loading; more difficult problems were successfully solved in the design of the connecting rod.

Mr. E. L. Diamond wrote in reply to the discussion and communications that he wished to express his thanks to all those who had contributed to what had become a comprehensive review of the subject. He could only refer individually to a few contributions, singled out because they appeared to require a specific answer.

In reading the discussion as a whole he felt that his brief remarks at the end of the meeting in London met the principal criticisms which had been made, but as the same points also figured in the written communications, a word or two of further explanation might help to remove the misapprehensions from which he believed they arose.

Mr. Carling had suggested that it was fundamentally no less correct to base a power formula on boiler dimensions and assume an engine of good design, than to base the power on the dimensions of the engine and assume a boiler of good design. Mr. Cox and Mr. Poulteney had stated categorically that it was essential to include the boiler in any consideration of locomotive power at speed. Both these contentions would be unanswerable if by "power" was meant "the maximum limit of power". The paper was, however, concerned with the power output at economical expansion ratios; this was the only practical basis for scheduling normal locomotive operation at speed, and the most logical basis for assessing design and performance of the engine, as distinct from the boiler. Possibly some colour was lent to the criticism because he had carried forward all the curves of mean effective pressure over the full range of speed, regardless of the fact that the boiler limit might thus be overstepped in some instances. To complete the information about the locomotives as a whole an additional curve representing the boiler limit would be necessary, as Mr. Cox and Mr. Poulteney had said. Such curves had been omitted because the necessary data were not in his possession, and would not affect the purpose of the paper, since the range of economic cut-offs at speed should always be well within the boiler limit of a properly designed locomotive. The point had been admirably brought out by Mr. Cardew in the fourth paragraph of his contribution.

No doubt it was also one of the diagrams in the paper, in this case Fig. 12, which had given rise to the mistaken idea that he was putting forward some kind of standard cycle like the "locomotive cycle" of Mr. Lawford Fry. Mr. Lawford Fry and Mr. Bond had both suggested that he had retracted the views he had put forward twenty years ago about such standard cycles, but that was not so. As Mr. Lawford Fry had quoted his somewhat jocular rejoinder on that occasion he might perhaps be permitted to quote the words in which he had expressed his real objection to the "locomotive cycle". "In the first place it varied for every diagram taken and was therefore no standard at all. Moreover it gave relative efficiencies that varied in a manner exactly opposite to the true variation." It was somewhat startling to him to find that he was understood to have put forward a modified "locomotive cycle" of his own, and to be attacked by some contributors for the same sound reasons.

He would therefore emphasize that what he had called the "basic diagram" was in no sense a standard: it was merely the means of calculating a particular engine's power from its dimensions, for a particular condition. It was, in fact, the individual diagram for that engine and that condition. The method was as old as the theory of the steam engine, and was precisely the same as was used by Dalby in 1905. It was in principle the normal method of calculating the power of any engine, but in the case of the locomotive there had always been the difficulty of determining the "diagram factor". The purpose of Fig. 12 was to show that thirty years ago the locomotive diagram factor would not have been a dependable basis for calculation, owing to extreme attenuation of the diagram at high speeds, whereas today the diagram approximated sufficiently closely to the theoretical diagram to justify the study presented in the first part of the paper for the purpose of determining the conditions necessary for economical running with different steam pressures, etc.

In the second part of the paper he had proposed a method taking account of the varying diagram factor, for any type of locomotive, by means of a single coefficient k. For convenience, in order that such a coefficient might be used for general comparisons, it was proposed, arbitrarily, that a standard exhaust pressure and a standard superheat temperature should be used for the basic diagrams. Objections had been raised to both of these, and he would therefore emphasize that other reasonable assumptions might be substituted; the method could be used equally well, but the numerical values of k might not then be strictly comparable with those given in the paper.

Mean piston speed would have appeared a more scientific basis for evaluation of the characteristic power coefficient k; on the other hand he was anxious not to give the coefficient a fictitious appearance of precision. A single coefficient was immensely more convenient for practical usage, although it could only be a compromise since it had to accommodate two different classes of loss, as he had pointed out in the paper. That being so he thought the convenience of working in terms of revolutions per minute was justified. If any abnormality of stroke had a marked influence on the power developed it would probably be apparent, as Mr. Holcroft had suggested, when the characteristic power equation was determined from the plotted points of mean effective pressure.

For the same reason (that they embraced complex elements) he made no claim that his curves represented a precisely true physical law of the variation of mean pressure throughout the whole range of speed; it was unlikely that any simple law of this type should do so, and it was noticeable that those who questioned the shape of his curves offered no agreed alternative. Deviations from the law suggested were shown and discussed in the paper on p. 415.

The feature which the suggested law *did* possess, however, was that it enabled the mean effective pressure in the high-speed range to be related with reasonable accuracy to the engine dimensions for all locomotives, which gave it a practical usefulness not possessed by curves of the type shown by Mr. Lawford Fry. This was also the answer to M. Chapelon's question as to the validity of drawing one curve through a theoretical point and a number of experimental points. M. Chapelon discussed the point of intersection of the curve with the zero speed axis as if it were to be regarded as the value for the diagram near zero speed, but he himself regarded it as the calculated or basic value, and he certainly attached no physical significance to the shape of the curves in the low-speed region. The function of the basic value was to supply the link required for calculating the actual values of the diagrams at speed from the engine dimensions, which was the whole purpose of the curves.

Mr. Lawford Fry's curves were for another reason not comparable with his own. The Pennsylvania test bulletins stated quite explicitly that the test designations gave the cut-off as indicated by the position of the reversing lever. The figures for cut-off which Mr. Fry had tabulated, were, to quote the words of the bulletins, "located by eye from examination of the indicator card". The report on the E6S locomotive included graphs of the deviation between the two measurements with the comment: "Irregularity in the difference is due in part to the difficulty of measuring the cut-off from either locomotive or from indicator diagram", and attributed the main difference to lack of rigidity in connexions between the reverse lever and the valve. As M. Chapelon and Mr. Carling had pointed out, however, indicator cards also involved lack of precision in the chain of connexions, and it was doubtful whether a more accurate relationship between mean effective pressure and speed could be established by disregarding the valve control mechanism in favour of the indicator diagram.

He fully accepted the fact that existing data were inaccurate, and that new methods would be necessary for the precise analysis of the complex phenomena occurring between steam chest and blast pipe. An empirical relationship of the kind suggested was the best that could be put forward at present, and, provided it gave results of the same order of accuracy as existing test methods, the main requirements were that it should be simple and practical. In normal service the reversing lever was the only means of determining the cut-off and all the diagrams in the paper were therefore based on the "cut-off" as registered by the lever.

Several contributors referred to the variation of k for the same engine at different cut-offs, and Mr. Cardew asked to what cut-off the curves in Fig. 20 applied. They were mean values for the range of economical cut-offs at speed for each locomotive. The remarkable thing was not that they varied, but that on the whole they were broadly consistent for each locomotive type. For the reasons already dwelt upon it was not to be expected that small variations in such a coefficient should have much significance, and he thought he was justified in taking mean values, with the reservations stated on p. 415 of the paper. Fig. 20 showed what could be expected, and he submitted that, for practical railway operation, the degree of accuracy provided was useful enough. The steam locomotive would always remain a somewhat temperamental machine whose behaviour could not be predicted with the same accuracy as that of an electric motor.

There was some discussion about the point of compression. He thought the second and third paragraphs on p. 410 of the paper fully covered the points raised by Mr. Lawford Fry and Mr. Bond. Mr. Marten and M. Chapelon, however, had raised a matter requiring much further investigation: the practical difficulty of associating maximum steam-flow area and minimum clearance volume.

He had not known that the Chapelon compound No. 4701 had so low a compression pressure in the low-pressure cylinders. From such indicator diagrams as were available he had inferred that compression was practically complete in the low-pressure cylinders of the French compounds, and that any loss of mean effective pressure due to the low-pressure clearance should be allowed to appear in the values of P_N (p. 412). He had had considerable correspondence with M. Chapelon as to the reason for the lower value of k for the later rebuild—at first apparent to neither. He thought the value of the treatment outlined in the paper was well shown by the fact that it had first called attention to this feature of the locomotive's performance, and then directed it to the true explanation, namely, the clearance volume.

It appeared from M. Chapelon's most valuable contribution that the adoption of a low compression pressure in the low-pressure cylinders might be an intentional method of offsetting the excessive clearance in the high-pressure cylinders, a good deal of the high-pressure clearance steam in effect going to fill up the low-pressure clearance space; M. Chapelon was justified in evaluating P_C in this way. As stated in the paper, "the method will give greater accuracy if the value [P_C] is calculated exactly for the point of compression appropriate to the particular valve gear design adopted". The points for No. 4701, as replotted in accordance with the new information supplied by M. Chapelon (Fig. 48), lay on the curves much more exactly than on the

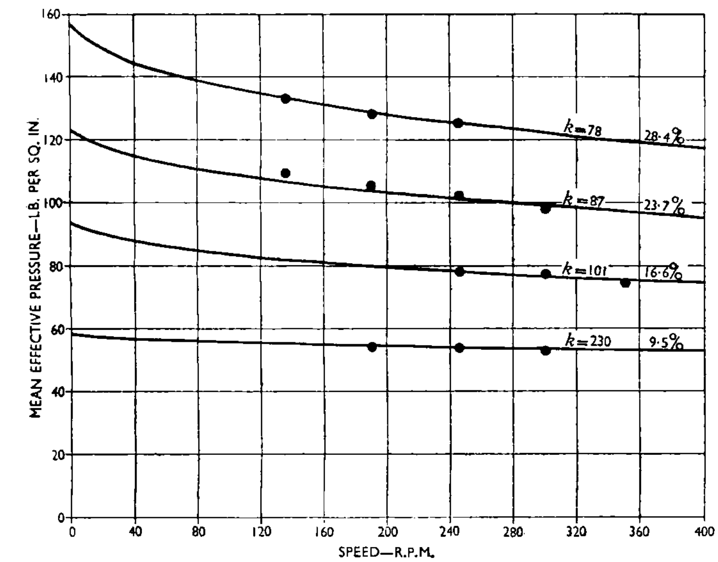

Fig. 48. Revised Mean Effective Pressure Curve for Chapelon Rebuilt Compound Locomotive No. 4701

Steam pressure, 284 lb. per sq. in. Characteristic power equation: $P_N = P_C\,(1-\frac{1}{89}\sqrt{N})$.

original curves of Fig. 15. Neglecting the abnormal value of k at the very short cut-off (in accordance with fifth paragraph, second column, p. 415), the mean characteristic coefficient k was 89, which placed this compound locomotive at the top in Fig. 20—a fact which he commended to the notice of Mr. Mills.

This effect of a small degree of low-pressure compression in reducing the equivalent clearance of a compound locomotive was a most interesting aspect of design, which Mr. Marten would probably note.

For normal locomotives there might be no objection to using the characteristic power coefficient to estimate approximately the power at speed at partial regulator openings, on the basis of the value of P_C at the reduced steam-chest pressures. Dr. Lomonossoff's comments threw some doubt on this however, and he himself would emphasize that all the values of P_N plotted in the paper were with full regulator opening.

He was greatly indebted to Mr. Carling for the diagrams with which he illustrated his remarks. It was gratifying that they supported the validity of curves of the type in the paper. Small variations of the factor k, however, especially at the shortest cut-offs where errors would be proportionately greatest, did not merit close study; the values were reasonably consistent for each locomotive, and mean values would be as reliable as most test results, and quite sufficiently so for practical purposes. Mr. Wilson's figures were too indirectly obtained to be useful in this connexion.

He would like to add that a most interesting discussion on the paper had taken place before the Western Branch of the Institution at Swindon (too late for full report in the PROCEEDINGS). In the course of this, Mr. Ell had stated that he had examined several series of indicator diagrams taken during tests of various Great Western locomotives on the testing plant at Swindon, and that remarkably consistent values for coefficient k had been found, which moreover were in close agreement with the figures given in the paper for modern locomotives with well-designed valve gears of conventional types.

Mr. Nock's question was answered in M. Chapelon's communication, but it would be apparent from the first part of the paper that, in order to realize an economy with full regulator opening, the locomotive must be designed to work efficiently at a sufficiently high expansion ratio.

Major-General Davidson's question as to the effect of pressure drop between boiler and steam chest was answered in the second paragraph of the right-hand column of p. 415, and it was well illustrated by the case of the K4S locomotive in its original form (Fig. 18, p. 414).

He was broadly in agreement with the remaining contributions, but all the considerations and qualifications advanced in the discussion could not have been covered in the brief length of the paper.